

Basic terms of automotive engineering

Basic terms of vehicle handling

Basic principles and systematics

Vehicle handling covers the dynamics of lateral, linear and vertical motion of passenger cars and commercial vehicles. Essentially, the dynamics of lateral motion deals with the steering behavior of the vehicle, the dynamics of linear motion with the acceleration behavior and braking response, and the dynamics of vertical motion with handling caused by excitations of road irregularities. The basic terms explained in the following (for quantities and units, see Table 1) describe vehicle handling in the important frequency range up to approx. 8 Hz. Many of these quantities are established in the German standard DIN 70000 [1] and the international standard ISO 8855 [2]. The quantities used can be subdivided into three groups.

Body

The motion of the body can generally be described as a rigid body. Even in the bodies of convertibles, deviations from rigid-body motion only occur above the frequency range of 10 to 15 Hz.

Wheel suspension

The front wheels have two degrees of freedom, i.e. the degree of freedom for compression and rebound and that for steering. The rear wheels generally only have the one degree of freedom for compression and rebound. In the case of vehicles with four-wheel steering, the rear wheels also have the degree of freedom for steering. These degrees of freedom are defined by the kinematics and elasto-kinematics of the axle. Kinematics of the axle is the pure rigid-body motion of the individual suspension arms, while elasto-kinematics describes the behavior of the axle in response to the effect of forces and moments.

Tires

The tire is the most important contact between the vehicle and its environment,

generating forces for propelling, braking and steering the vehicle in interaction with the road.

Table 1: Quantities and units

Quantity	Unit
α	Slip angle
β	Float angle
δ_H	Steering-wheel angle
δ_R	Toe angle of right wheel
δ_L	Toe angle of left wheel
δ_A	Axle steering angle (steering angle)
ε_V	Longitudinal inclination at wheel center
ε_{BV}	Anti-dive angle
ε_{AV}	Anti-lift angle
ψ	Yaw angle
φ	Roll angle
θ	Pitch angle
γ	Camber angle
σ	Kingpin angle
τ	Caster angle
λ	Tire slip
λ_B	Braking-force distribution
ω_R	Wheel angular speed
a_x	Longitudinal acceleration
a_y	Lateral acceleration
a_z	Vertical acceleration
a_t	Tangential acceleration
a_c	Centripetal acceleration
F_S	Lateral force
F_L	Longitudinal force
F_Z	Axle load
h	Height of center of gravity
h_W	Height of roll pole
i_s	Steering ratio
l	Wheelbase
M_H	Steering-wheel torque
M_R	Tire aligning torque
n_τ	Caster offset
n_v	Caster offset in wheel center
n_R	Tire caster offset
r_σ	Kingpin offset at wheel center
r_{st}	Deflection-force lever arm
r_l	Scrub radius
r_{dyn}	Dynamic rolling radius
s	Track width
v_x	Longitudinal velocity
v_y	Lateral velocity
v_z	Vertical velocity
v_{RAP}	Speed at wheel contact point
v_{RMP}	Speed at wheel center
X_A	Starting-torque compensation
X_B	Braking-torque compensation

Basic terms for the body*Translational (uniform) motion*

The body is described by three translational and three rotational degrees of freedom (Figure 1). Generally, the system of coordinates (rectangular right-handed system) has its origin at the vehicle's center of gravity. The x -axis points forwards, i.e. in the direction of travel. It is situated in the plane that is perpendicular to the road surface. This plane is called the vehicle center plane. Situated perpendicular to the vehicle center plane is the y -axis, which points, when viewed in the direction of travel, to the left. The z -axis points upwards. The translational motions are designated as follows:

- In x -direction: linear motion.
- In y -direction: lateral motion.
- In z -direction: lift.

Translational velocities and accelerations

Single time derivation of the translational motion variables determines the linear, lateral and vertical velocities (v_x , v_y , v_z). Renewed time derivation produces the linear acceleration a_x , lateral acceleration a_y and vertical acceleration a_z .

Float angle

In the dynamics of lateral motion the center of gravity of the body does not always move along the x -axis. The angle that forms between the vehicle center plane and the trajectory is called the float angle β (Figure 2). It is counted from the

vehicle center plane to the trajectory. The float angle is calculated from the linear velocity v_x and the lateral velocity v_y :

$$\beta = \arctan \frac{v_y}{v_x}$$

Because linear velocity during forward driving is by definition positive, the sign preceding the lateral velocity in this case also determines the sign preceding the float angle.

Centripetal acceleration and tangential acceleration

Acceleration is proportioned in the horizontal plane along the instantaneous vehicle trajectory into centripetal acceleration a_c and tangential acceleration a_t . The centripetal acceleration is that part which is perpendicular to the trajectory, while the tangential acceleration is that part in the tangential direction of the trajectory.

Yaw, pitch and roll

The following terms have been established to cover rotational motion:

- pure rotation about the z -axis is called yaw and is described by ψ ,
- pure rotation about the y -axis is called pitch and is described by θ ,
- pure rotation about the x -axis is called roll and is described by ϕ .

The signs preceding these rotations correspond to those of a rectangular right-handed system. The counting directions

Figure 1: Translational and rotational degrees of freedom of the body

ψ Yaw angle,
 ϕ Roll angle,
 θ Pitch angle,
 S Center
 of gravity.

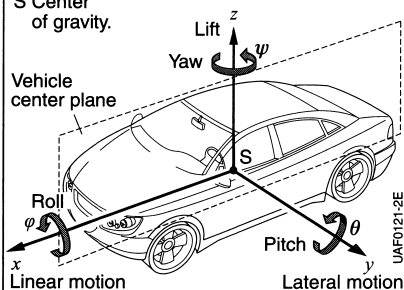
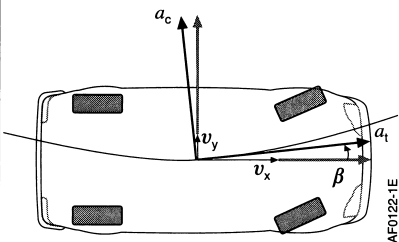


Figure 2: Float angle, centripetal and tangential acceleration

β Float angle,
 a_t Tangential acceleration,
 a_c Centripetal acceleration,
 v_x Longitudinal velocity,
 v_y Lateral velocity.



are shown in Figure 1. In many important driving maneuvers the body simultaneously moves about more than one axis. Because, when viewed mathematically, the three rotations are not commutative, the following order for the rotations has been defined in DIN 70000:

1. Yaw,
2. Pitch,
3. Roll.

Rotational velocities and accelerations

Single time derivation of the rotational motion variables produces the yaw, pitch and roll velocities. Renewed time derivation produces the yaw, pitch and roll accelerations.

Yaw moment, pitch moment, rolling moment

External forces acting on the vehicle can generate a moment about the center of gravity. This moment is broken down into three components. The portion of moment along the z -axis is called yaw moment, that along the y -axis pitch moment, and along x -axis rolling moment.

Measured variables

In the interests of vehicle development it is sensible to use a special measurement to enable vehicle handling to be measured with greater accuracy. In the dynamics of lateral motion the translational accelerations and the attitude angles are often measured by gyro-stabilized platforms. The absolute positions are recorded with GPS measuring systems. The linear and lateral velocities are each measured with proximity-type velocity sensors for the float angle.

In the dynamics of vertical motion the translational accelerations are measured in each of the three directions in space at different points of the body. From these the most important body accelerations can be determined, i.e. those for lift, pitch and roll.

Basic terms for the wheel suspension

Steering-wheel angle and steering-wheel torque

The steering-wheel angle δ_H is the turning angle of the steering wheel measured from the straight-ahead position. The angle is positive for a left-hand curve.

The driver must apply a torque when adjusting the steering-wheel angle. This torque is called the steering-wheel torque M_H and is also positive for a left-hand curve.

Toe angle

Moving the steering wheel causes the two front wheels to turn in the same direction. This alters the toe angle of the right and left wheels (δ_R and δ_L) respectively. The toe angle is the angle between the vehicle center plane and the wheel center plane when projected onto the road surface (Figure 3). In the case of a positive turn, i.e. in the counterclockwise direction about the wheel's z -axis, the toe angle is also positive.

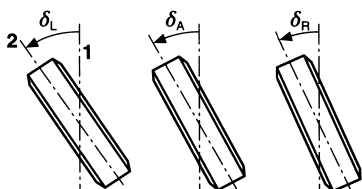
On vehicles with four-wheel steering the rear wheels also move.

Axle steering angle, steering angle

When the steering wheel is moved, different toe angles are obtained at the two front wheels. The difference between the toe angles can amount to several degrees with the steering-wheel angle at maximum. When viewed geometrically, the toe angle of the inside wheel is greater than the toe of the outside wheel. The mean

Figure 3: Toe angle and axle steering angle

- 1 Vehicle center plane,
 - 2 Wheel center plane.
- δ_L Toe angle of left wheel,
 δ_R Toe angle of right wheel,
 δ_A Axle steering angle (steering angle),
 mean toe angle.



toe angle is called the axle steering angle δ_A or simply the steering angle (Figure 3).

Steering ratio

Essentially, on account of the steering gear but also on account of the kinematics of the front axle, the toe angle obtained at the wheels is significantly smaller than the steering-wheel angle. In such a case where there are no acting forces or torques and where the vehicle is barely loaded, the steering ratio i_s is governed by the following equation:

$$i_s = \frac{2 \delta_H}{(\delta_L + \delta_R)}.$$

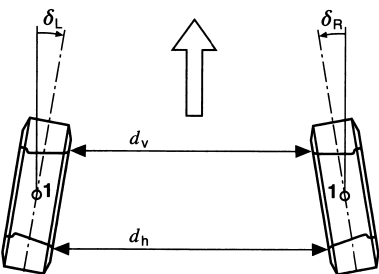
Toe-in and toe-out

When the steering wheel is in the straight-ahead position, the toe angles of the front wheels are in the dynamically advantageous range of 0.1 to 0.3°. When the distance of the rim flanges ahead of the wheel centers is smaller than that of the rim flanges after the wheel centers, this is referred to as wheel toe-in (Figure 4). When the situation is reversed, this is referred to as wheel toe-out. Both toe-in and toe-out are given in degrees (°).

The terms toe-in and toe-out are also used to refer to a single wheel. In this case, toe-in means that the wheel has a toe angle in the direction of the vehicle center plane and toe-out that it has a toe angle against the direction of the vehicle center plane.

Figure 4: Toe-in

- 1 Wheel center.
- d_v Distance between rim flanges, front,
- d_h Distance between rim flanges, rear,
- δ_L Toe angle of left wheel,
- δ_R Toe angle of right wheel,
- ▷ Direction of travel.



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Camber angle

The camber angle γ is the angle between the vehicle center plane and the wheel center plane when projected onto the z - y plane. The camber angle is positive when the wheels are further away from the vehicle center plane at the top than at the bottom (Figure 5a). On account of the axle kinematics the camber angle relative to the body is dependent on the spring travel.

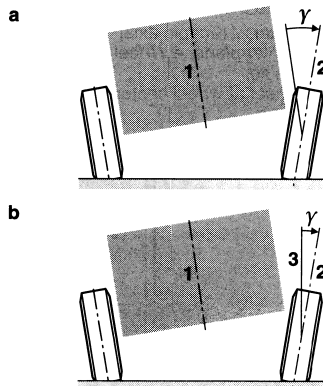
Further to this definition, the camber angle relative to the road is likewise important to vehicle handling. The camber angle relative to the road is the angle between the wheel center plane and the normal line to the road surface (Figure 5b). The preceding sign is determined according to the rectangular right-handed system. If the vehicle center plane is perpendicular to the road surface, both definitions of the camber angle are equal in terms of amount. Otherwise, it is important to observe the precise definition.

Steering axis, kingpin axis

During steering the wheels move not about their z -axis, but rather about the steering axis, also known as the kingpin axis. The position of the kingpin axis is

Figure 5: Camber

- a) Relative to the body,
- b) Relative to the road.
- 1 Vehicle center plane,
- 2 Wheel center plane,
- 3 Normal line to the road surface.
- γ Camber angle.



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essentially determined by the kinematics of the axle (Figure 6 and Figure 7).

Caster angle, caster offset

in wheel center and caster offset

When the wheel and the kingpin axis are projected onto the vehicle center plane, the kingpin axis is inclined by the caster angle τ . It is positive when the top end of the kingpin axis is inclined towards the rear (Figure 6). In this projection the kingpin axis generally does not pass through the wheel center; instead, it is offset by the caster offset in wheel center n_v to the rear. The distance between the wheel contact point and the point where the kingpin axis intersects the road surface is called the caster offset n_r . If this point of intersec-

tion is located in front of the wheel center (Figure 6), the caster offset is positive. Caster offsets are typically in the range of 15 to 30 mm.

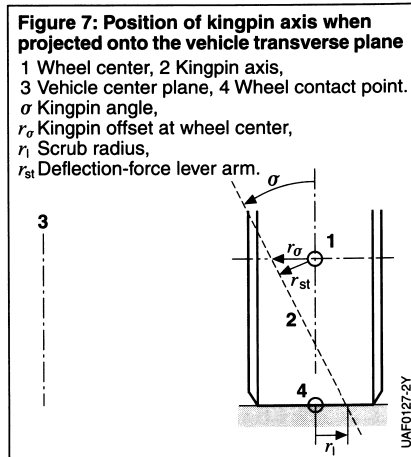
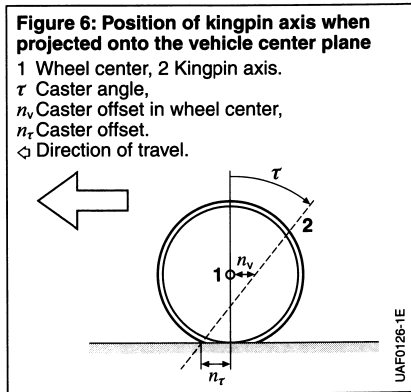
Kingpin angle, kingpin offset at wheel center, deflection-force lever arm and scrub radius

When the wheel and the kingpin axis are projected onto the vehicle transverse plane, the kingpin axis is inclined by the kingpin angle σ (Figure 7). The kingpin angle (also known as the kingpin inclination) is positive when the kingpin axis is inclined to the vehicle center. Kingpin angles are generally positive.

The distance between the wheel center and the kingpin axis parallel to the road is called the kingpin offset at wheel center r_o . The kingpin offset at wheel center is positive when the wheel center is further from the vehicle center plane than the kingpin axis (Figure 7). The shortest connection between the wheel center and the kingpin axis is called the deflection-force lever arm r_{st} . The deflection-force lever arm is positive when the wheel center is further from the vehicle center plane than the kingpin axis.

The axes are arranged in such a way that the kingpin offset at wheel center and the deflection-force lever arm are as small as possible. This arrangement prevents deflection forces from being generated in the steering.

The distance between the wheel contact point and the point where the kingpin axis intersects the road surface is called the scrub radius r_i (Figure 7). The scrub radius is positive when the wheel contact point is further from the center plane than the kingpin axis. When the scrub radius is positive, the wheel moves towards toe-out under braking forces. This behavior is particularly advantageous during the "braking in a curve" maneuver. When the scrub radius is negative, the wheel moves towards toe-in under braking forces. During the " μ -split braking" maneuver with different coefficients of friction on the left and right sides of the vehicle this axle behavior creates the precondition for more stable vehicle handling. On account of these different effects the scrub radius is configured to be as small as possible.



Transverse pole, roll pole, roll axis

When the axle is subjected to compression and rebound the position of the tires is determined primarily by the kinematics and elastokinematics. The tire moves transversally to the direction of travel about the transverse pole (Figure 8). The speeds for example at the wheel contact point (v_{RAP}) and at the wheel center (v_{RMP}) are during compression and rebound perpendicular on the connecting line to the transverse pole. The position of the transverse pole changes during compression and rebound.

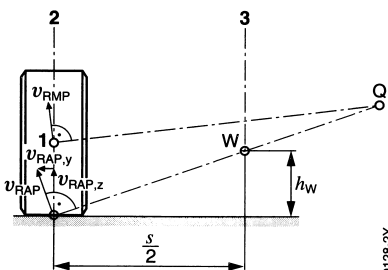
The vehicle body moves in the case of low lateral accelerations about the roll pole of the respective axle (Figure 8). The roll pole is on the connecting line for wheel contact point and transverse pole in the vehicle center plane, i.e. at half the track width ($s/2$). The height h_w of the roll pole can thus be easily calculated:

$$h_w = \frac{v_{RAPy}}{v_{RAPx}} \frac{s}{2}.$$

The height of the roll pole is typically below 120 mm. In order in the case of high lateral accelerations to avoid the support effect, i.e. the jacking effect, the height of the roll pole decreases with compression. The roll pole is also known as the roll center or instantaneous center.

Figure 8: Transverse pole and roll pole

- 1 Wheel center,
 - 2 Tire center plane,
 - 3 Vehicle center plane.
- Q Transverse pole, W Roll pole.
 s Track width, h_w Height of roll pole.
 v_{RAP} Speed at wheel contact point,
 v_{RMP} Speed at wheel center.



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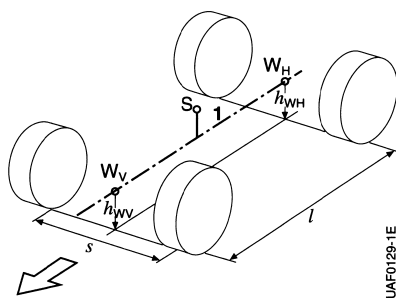
The connection between the roll pole of the front axle and the roll pole of the rear axle is called the roll axis (Figure 9). The center of gravity of the body is usually above the roll axis. Center-of-gravity heights for sedans range between 550 and 650 mm. This roll axis applies for lower lateral accelerations. In the case of higher lateral accelerations, both the suspension adjustment and the axle behavior must be taken into consideration. The roll axis is then not inevitably in the vehicle center plane.

Longitudinal pole, pitch pole, pitch axis, braking-torque compensation, and starting-torque compensation

When the spring movement of the axle is projected onto the vehicle center plane, the tire moves about the longitudinal pole L (Figure 10). During compression the wheel center moves upwards at the longitudinal inclination at wheel center ε_v . The speeds at the wheel contact point (v_{RAP}) and the wheel center (v_{RMP}) are during compression and rebound perpendicular on the connecting line to the longitudinal pole. The position of the longitudinal pole can change during compression and rebound. The angle between the connection of the wheel contact point to the longitudinal pole and the road is called

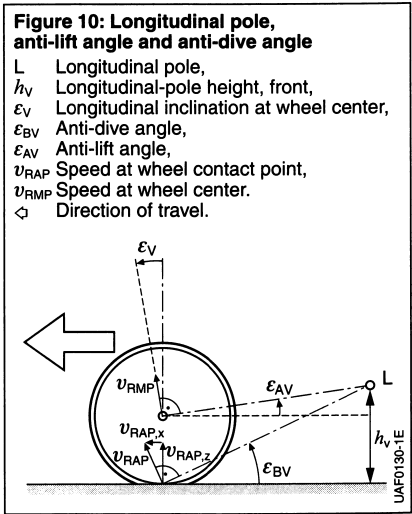
Figure 9: Roll axis

- 1 Roll axis.
- S_o Center of gravity,
 W_v Roll pole, front axle,
 W_H Roll pole, rear axle,
 s Track width,
 l Wheelbase,
 h_{wv} Roll-pole height, front axle,
 h_{wH} Roll-pole height, rear axle.
 $\diamond$ Direction of travel.



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the anti-dive angle ε_{BV} . The angle between the connection of the wheel center to the longitudinal pole and the parallel line to the road is called the anti-lift angle ε_{AV} .
The longitudinal pole of the front axle is behind the front wheels and the longitudinal pole of the rear axle is in front of the rear wheels (Figure 11).

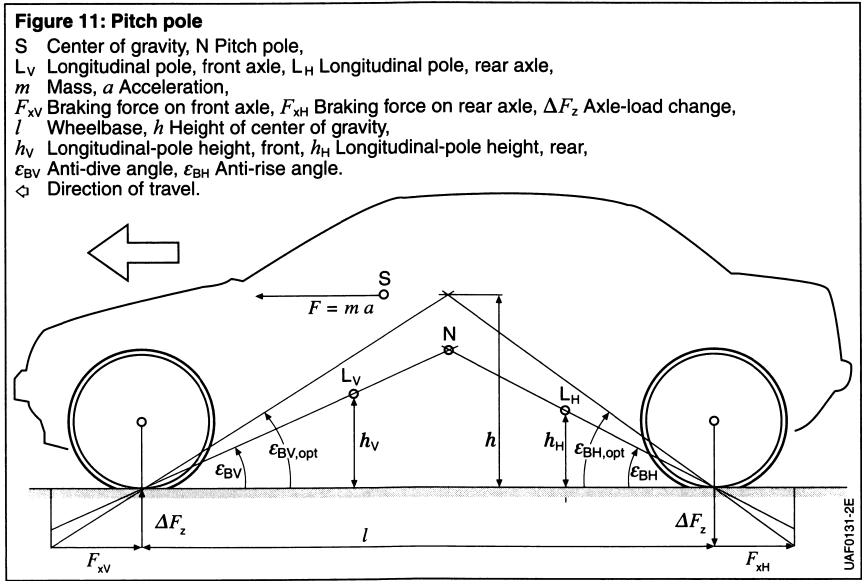


When the vehicle is braked, the front axle load is increased by ΔF_z and the rear axle load decreased by ΔF_z . At the vehicle center of gravity the force $F = m a$ is applied. With braking-force distribution λ_B the braking forces $F_{xV} = \lambda_B F_x$ and $F_{xH} = (1 - \lambda_B) F_x$ are applied to the front and rear axles respectively. In the optimum case where the resulting force from F_{xV} and ΔF_z on the front axle passes exactly through the longitudinal pole and similarly on the rear axle, no spring movement occurs in the vehicle's body springs. For the optimum anti-dive and anti-rise angles:

$$\tan(\varepsilon_{BV,opt}) = \frac{1}{\lambda_B} \frac{h}{l},$$
$$\tan(\varepsilon_{BH,opt}) = \frac{1}{(1-\lambda_B)} \frac{h}{l},$$

with wheelbase l and height of center of gravity h .

The two connecting lines between the wheel contact points and the longitudinal pole intersect at the pitch pole N (Figure 11). The pitch pole N is always below the height of the center of gravity so that an expected vehicle pitching motion towards the rear can always be effected



when braking. The pitch axis passes through the pitch pole and is perpendicular to the vehicle center plane. Braking-torque compensation X_{BV} and X_{BH} is a measure of how effectively optimum braking compensation has been implemented. The following applies:

$$X_{BV} = \frac{\tan(\varepsilon_{BV})}{\tan(\varepsilon_{BV,opt})} 100\%,$$

$$X_{BH} = \frac{\tan(\varepsilon_{BH})}{\tan(\varepsilon_{BH,opt})} 100\%.$$

The same applies to acceleration. It is necessary to take into account firstly the type of drive, i.e. four-, front- or rear-wheel drive, and secondly the fact that the tractive force is applied at the wheel center. In the general case of four-wheel drive the following applies to starting-torque compensation X_{AV} and X_{AH} :

$$X_{AV} = \frac{\tan(\varepsilon_{AV})}{\tan(\varepsilon_{AV,opt})} 100\%,$$

$$X_{AH} = \frac{\tan(\varepsilon_{AH})}{\tan(\varepsilon_{AH,opt})} 100\%.$$

Measured variables

Steering-wheel angle and steering-wheel torque are measured using special measuring steering wheels. If the measuring accuracy is sufficient, the steering-wheel-angle sensor fitted as standard in many vehicles can also be used.

Special measuring devices can be used in both mobile and stationary applications to measure toe and camber angles. These angles are typically measured on special test benches.

Special test benches are also used to measure the axle steering angle and the steering ratio.

The variables for the position of the kingpin axis are generally not measured directly. The axle pivot points are often recorded by means of geometric measurements and the following variables are calculated from them: caster angle, caster offset in wheel center, caster offset, kingpin angle, kingpin offset at wheel center, deflection-force lever arm, and scrub radius. The roll pole can be determined by measuring the track change during the reciprocal compression and rebound of an axle. The roll axis is obtained from the roll poles of the front and rear axles.

The pitch pole and starting- and braking-torque compensation are usually not measured directly, but rather determined from the measured kinematic points of the axle. The same applies to the longitudinal inclination at wheel center, and the anti-lift/anti-squat and anti-dive/anti-rise angles.

Basic terms for tires

The most important external forces and moments which act on a vehicle occur during the transmission of force between tire and road surface. Added to this are the wind forces which act on the vehicle special situations.

Tire contact area

Force is transmitted between tire and road surface by friction in the contact area. The two most important types of friction are adhesive friction (intermolecular adhesive force) and hysteretic friction (gearing force).

Lateral force, longitudinal force

Forces can be generated in the road plane by friction. Lateral force F_S is the force perpendicular to the wheel center plane; longitudinal force F_U runs in the direction of the wheel center plane (Figure 12). The forces are generally not applied exactly at the wheel contact point such that moments are likewise generated in relation to the wheel contact point (see Tire aligning torque).

Figure 12: Lateral force and longitudinal force

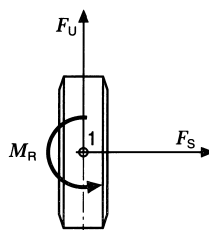
(View from above).

1 Wheel contact point.

F_U Longitudinal force,

F_S Lateral force,

M_R Tire aligning torque.



Wheel load and slip angle

When lateral and longitudinal forces occur simultaneously, the forces may be mutually influenced. The situation where lateral and longitudinal forces occur but not combined is considered in the following. The generated lateral force F_S is dependent on the wheel load and the slip angle α . A dependence of lateral force F_S on speed can generally be ignored. The wheel load is the force with which the wheel center is pressed towards the road-surface plane. The slip angle α is the angle between the direction of motion of the wheel contact point and the wheel center plane (Figure 13).

When the wheel load is kept constant and the slip angle α is increased, the lateral force initially increases linearly. The lateral force reaches its maximum level at a slip angle of approx. 5° and then decreases slightly (Figure 14).

Tire aligning torque and tire caster offset

With small slip angles the lateral force is applied after the wheel contact point. When the slip angle is increased, the lateral force moves increasingly towards the wheel contact point and can also be located ahead of the wheel contact point. The distance between the lateral application point and the wheel contact point is called the tire caster offset n_R . The lat-

eral force therefore generates a moment about the tire's vertical axis, the tire aligning torque M_R :

$$M_R = F_S \cdot n_R .$$

This produces, at constant wheel load, a curve as shown in Figure 15 for the tire aligning torque.

If the tire aligning torque is positive, it helps to make the slip angle smaller in terms of amount. This behavior helps to make the turned wheels return to the straight-ahead position when the steering wheel is released.

Slip and rolling radius

Similarly to the slip angle α for lateral force F_S , slip λ is the variable which at constant wheel load determines longitudinal force F_U . Slip occurs when the speed v_{xR} at which the wheel center moves in the longitudinal direction differs from the speed v_U at which the circumference rolls. The circumferential speed is calculated from the angular speed ω_R of the wheel and the dynamic rolling radius r_{dyn} :

$$v_U = \omega_R \cdot r_{dyn} .$$

A distinction is made between static and dynamic rolling radius. The static rolling

Figure 13: Slip angle
(View from above).
 F_S Lateral force,
 M_R Tire aligning torque,
 α Slip angle.
R Wheel contact point.

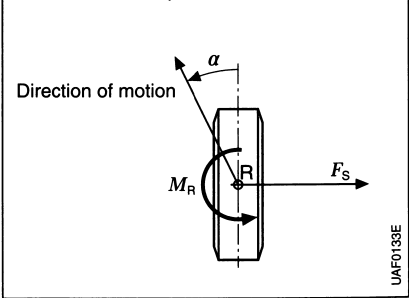
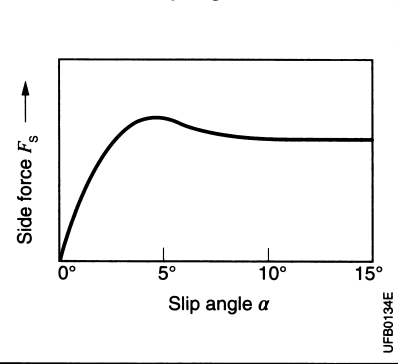


Figure 14: Lateral force at constant wheel load as a function of slip angle



radius is the shortest distance between wheel center and tire contact area. The dynamic rolling radius r_{dyn} is calculated by means of the circumference U :

$$r_{\text{dyn}} = \frac{U}{2\pi}.$$

Slip λ_A for tractive forces is defined as:

$$\lambda_A = \frac{\omega_R r_{\text{dyn}} - v_{xR}}{\omega_R r_{\text{dyn}}}.$$

Similarly, slip λ_B for braking forces is defined as:

$$\lambda_B = \frac{\omega_R r_{\text{dyn}} - v_{xR}}{v_{xR}}.$$

According to this definition, drive slip is always positive and brake slip always negative. These two slip definitions ensure that when a wheel is locked ($\omega_R = 0$, $\lambda_B = -1$) a slip of -100% is obtained and when a wheel is spinning ($v_{xR} = 0$, $\lambda_A = 1$) a slip of 100% is obtained.

If the drive slip is increased at constant wheel load, the tractive force (longitudinal force) increases linearly. At approx. 10% drive slip the longitudinal force reaches its maximum level and then falls again (Figure 16). The same applies to brake slip. Here the maximum braking force is created at approx. -10% .

Measured variables

The slip angle α is measured, along similar lines to the float angle, with two proximity-type speed sensors. The wheel speed and the longitudinal velocity are measured for slip λ . The dynamic rolling radius r_{dyn} is determined on test benches.

The tire aligning torque M_R , the lateral force F_S , and the longitudinal force F_U can be recorded in mobile operation by multi-component measuring wheels. Because this is very costly, tire forces and tire moments are generally measured on stationary test benches or determined using special vehicles directly on the road. Up to now, precise measurement of tire forces and tire moments has been accompanied by many systematic failures.

References

- [1] DIN 70000 (earlier standard) Road vehicles – Vehicle dynamics and road-holding ability – Vocabulary.
- [2] ISO 8855: Road vehicles – Vehicle dynamics and road-holding ability – Vocabulary.
- [3] B. Heissing, M. Ersoy (Editors): Fahrwerkhandbuch. Vieweg+Teubner Verlag, 2008.

Figure 15: Tire aligning torque at constant wheel load as a function of slip angle

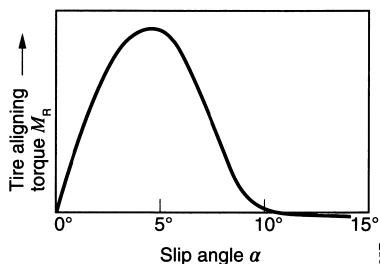
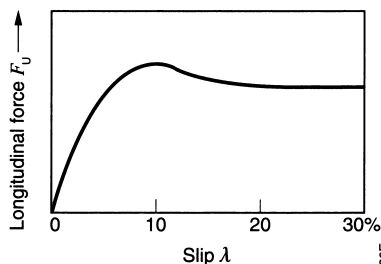


Figure 16: Longitudinal force at constant wheel load as a function of drive slip



Motor-vehicle dynamics

The dynamics of motor vehicles, also called driving dynamics, comprises the motion of the vehicle about the three main axes and is divided into longitudinal dynamics (motion towards the longitudinal axis), transverse dynamics (motion towards the transverse axis), and vertical dynamics (motion about the vertical axis). Vertical dynamics plays a central role in driving-dynamics control.

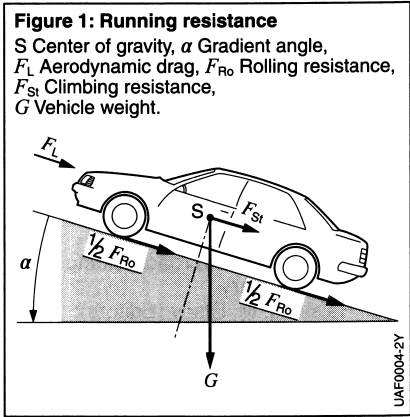


Table 1: Quantities and units

Quantity	Unit
A Largest cross-section of vehicle	m^2
a Acceleration, braking (deceleration)	m/s^2
c_w Drag coefficient	—
F Motive force	N
F_{cf} Centrifugal force	N
F_K Cornering resistance	N
F_L Aerodynamic drag	N
F_{Ro} Rolling resistance	N
F_{St} Climbing resistance	N
F_W Running resistance	N
f_K Coefficient of cornering resistance	—
f_R Coefficient of rolling resistance	—
G Force due to weight, $G = m g$	N
G_B Sum of wheel forces on driven or braked wheels	N
g Gravitational acceleration $\approx 9.81 \text{ m/s}^2$	m/s^2

Dynamics of linear motion

The dynamics of linear motion (longitudinal dynamics) comprises the running resistances, the acceleration and deceleration, the forces and power outputs for drive and deceleration operations, and the energy expenditure required for these purposes, which is referred to as consumption.

Total running resistance

The running resistance F_W is the sum total of rolling resistance F_{Ro} , aerodynamic drag F_L , and climbing resistance F_{St} (Figure 1):

$F_W = F_{Ro} + F_L + F_{St} .$

The power which must be transmitted through the drive wheels to overcome running resistance (running-resistance power) is:

$P_W = F_W v$ (quantity equation) or

$P_W = \frac{F_W v}{3600}$ (numerical-value equation).

Numerical-value equation with P_W in kW, F_W in N, v in km/h.

Quantity	Unit
i Gear or transmission ratio between engine and drive wheels	—
k_m Rational inertia coefficient	—
M Engine torque	Nm
m Vehicle mass	kg
n Engine speed	min^{-1}
P Power	W
P_W Motive power	W
p Gradient ($\tan \alpha$)	%
r Dynamic tire radius	m
s Distance traveled	m
t Time	s
v Driving speed	m/s
v_0 Headwind speed	m/s
W Work	J
α Incline angle, Gradient angle	$^\circ$
μ_r Coefficient of static friction	—
η Efficiency	—

Additional symbols and units in text.

Rolling resistance

The rolling resistance F_{R0} is the product of deformation processes which occur at the contact patch between tire and road surface. The following applies:

$$F_{R0} = f_R G \cos \alpha = f_R m g \cos \alpha.$$

An approximate calculation of the rolling resistance can be made using the coefficients provided in Table 2 and Figure 2.

The increase in the coefficient of rolling resistance f_R is directly proportional to the level of deformation, and inversely proportional to the radius of the tire. The coefficient will thus increase in response to greater loads, higher speeds and lower tire inflation pressure.

In turns, the rolling resistance is augmented by the cornering resistance.

$$F_K = f_K G.$$

The coefficient of cornering resistance f_K is a function of driving speed, curve radius, suspension geometry, tires, tire inflation pressure, and the vehicle's response under lateral acceleration.

Aerodynamic drag

The aerodynamic drag is calculated as a function of air density ρ , cross-sectional area A , drag coefficient c_w , and the square of speed v as:

$$F_L = 0.5 \rho c_w A (v + v_0)^2 \text{ or}$$

$$F_L = 0.0386 \rho c_w A (v + v_0)^2$$

with v in km/h, F_L in N, ρ in kg/m³, A in m², air density ρ ($\rho = 1.202$ kg/m³ at 200 m altitude).

Typical value ranges for drag coefficient and cross-sectional area are listed in Table 3 for different vehicle body configurations.

The aerodynamic drag is

$$P_L = F_L v = 0.5 \rho c_w A v (v + v_0)^2$$

or

$$P_L = 12.9 \cdot 10^{-6} c_w A v (v + v_0)^2$$

with P_L in kW, F_L in N, v and v_0 in km/h, A in m², $\rho = 1.202$ kg/m³. The largest cross-section of an automobile is:

$$A \approx 0.9 \times \text{lane width} \times \text{height}.$$

Empirical determination of coefficients for aerodynamic drag and rolling resistance

The vehicle is allowed to coast down in neutral under windless conditions on a level road surface. The time that elapses while the vehicle coasts down by a specific increment of speed is measured from two initial velocities, v_1 (high speed) and

Table 2: Coefficients of rolling resistance

Road surface	Coefficient of rolling resistance f_R
Pneumatic car tires on	
large set pavement	0.011
small set pavement	0.011
concrete, asphalt	0.008
rolled gravel	0.02
unpaved road	0.05
field	0.1 - 0.35
Pneumatic truck tires on	
concrete, asphalt	0.006 - 0.01
Strake wheels in	
field	0.14 - 0.24
Track-type tractor in	
field	0.07 - 0.12
Wheel on rail	0.001 - 0.002

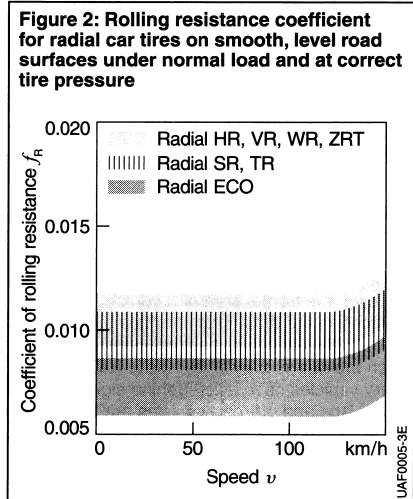
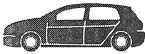
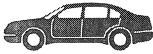
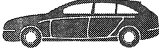
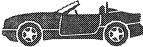
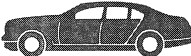


Table 3: Drag coefficients and aerodynamic drag for various vehicle body configurations

		Drag coefficient c_W	Frontal area in m^2	Aerodynamic drag in kW at a driving speed of	
				40 km/h	120 km/h
	Subcompact	0.29 to 0.37	2.05 to 2.20	0.5 to 0.7	13.2 to 18.1
	Compact class	0.22 to 0.32	2.18 to 2.28	0.4 to 0.6	10.7 to 16.2
	Medium class	0.23 to 0.35	2.20 to 2.38	0.4 to 0.7	11.3 to 18.5
	Station wagon	0.27 to 0.35	2.20 to 2.38	0.5 to 0.7	13.2 to 18.5
	Van	0.25 to 0.35	2.40 to 3.20	0.5 to 0.9	13.4 to 24.9
	Convertible				
	- closed	0.28 to 0.38	1.94 to 2.20	0.4 to 0.7	12.1 to 18.6
	- open	0.35 to 0.50	1.84 to 2.10	0.5 to 0.9	14.3 to 23.4
	Off-road vehicle	0.29 to 0.55	2.49 to 3.15	0.6 to 1.3	16.1 to 38.6
	Sports cars	0.27 to 0.40	1.65 to 2.20	0.4 to 0.7	10.0 to 19.6
	Luxury class	0.23 to 0.35	2.28 to 2.65	0.4 to 0.8	11.7 to 20.6

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Table information:

The midsize class market segment features the lowest c_W values, followed by the luxury class, compact class, sports cars, closed convertibles, station wagons, vans, small cars up to the highest values for open-top convertibles and off-road vehicles.

However, the frontal area of a vehicle also has a significant influence on the power requirement for overcoming the drag. This changes the order with regard to the c_W value. Midsize vehicles also require the lowest power here, followed by the compact class, sports cars and closed-top convertibles, which are nearly the same. This is followed by small vehicles and station wagons. Due to the size, luxury vehicles have fallen behind significantly and then open-top convertibles due to their high c_W value. Vans and off-road vehicles are found at the end.

This also shows that the aerodynamic drag at 40 km/h hardly differs between the classes. At 120 km/h, however, there are substantial differences because the power requirement increases with the speed cubed.

v_2 (low speed). This information is used to calculate the mean deceleration rates a_1 and a_2 . The formulae and example from Table 4 are based on a vehicle weighing $m = 1450 \text{ kg}$ with a cross-section $A = 2.2 \text{ m}^2$.

The method is suitable for application at driving speeds of less than 100 km/h .

Climbing resistance and downgrade force

Climbing resistance (F_{St} with positive sign) and downgrade force (F_{St} with negative sign) are calculated as:

$$F_{St} = G \sin \alpha = m g \sin \alpha$$

or, for a working approximation:

$$F_{St} \approx \frac{m g p}{100 \%}$$

These equations apply to gradients up to $p \leq 20 \%$, because at small angles the following applies:

$$\sin \alpha \approx \tan \alpha \text{ (less than } 2 \% \text{ error).}$$

Refer to Table 5 for the climbing resistance, normalized to 1000 kg vehicle mass, as a function of the gradient p and the gradient angle α .

“in” in the gradient scale in Table 5 means “divided by” and is customary in English-speaking countries.

Table 5: Gradient angle and climbing resistance

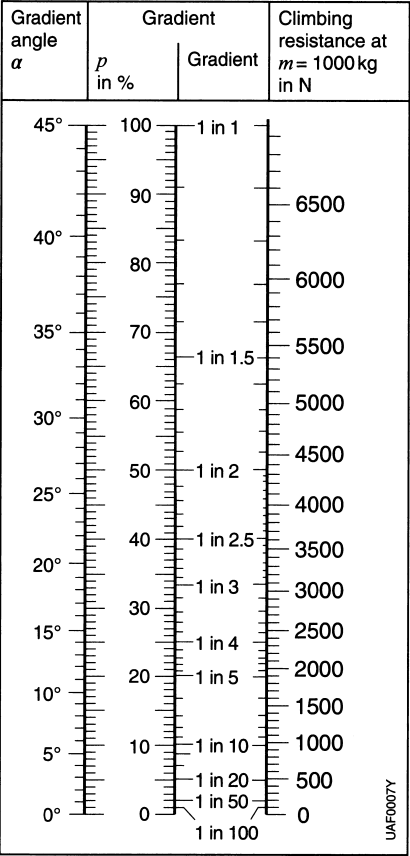


Table 4: Empirical determination of coefficients for aerodynamic drag and rolling resistance

	1st trial (high speed)	2nd trial (low speed)
Initial velocity	$v_{a1} = 60 \text{ km/h}$	$v_{a2} = 15 \text{ km/h}$
Terminal velocity	$v_{b1} = 55 \text{ km/h}$	$v_{b2} = 10 \text{ km/h}$
Interval between v_a and v_b	$t_1 = 7.8 \text{ s}$	$t_2 = 12.2 \text{ s}$
Mean velocity	$v_1 = \frac{v_{a1} + v_{b1}}{2} = 57.5 \text{ km/h}$	$v_2 = \frac{v_{a2} + v_{b2}}{2} = 12.5 \text{ km/h}$
Mean deceleration	$a_1 = \frac{v_{a1} - v_{b1}}{t_1} = 0.64 \frac{\text{km/h}}{\text{s}}$	$a_2 = \frac{v_{a2} - v_{b2}}{t_2} = 0.41 \frac{\text{km/h}}{\text{s}}$
Drag coefficient (all quantities as numerical values)	$c_w = \frac{6m(a_1 - a_2)}{A(v_1^2 - v_2^2)} = 0.29 \text{ (with } m = 1450 \text{ kg, } A = 2.2 \text{ m}^2\text{)}$	
Coefficient of rolling resistance (all quantities as numerical values)	$f_R = \frac{28.2(a_2 v_1^2 - a_1 v_2^2)}{10^3 \cdot (v_1^2 - v_2^2)} = 0.011$	

Climbing power is calculated as:

$$P_{St} = F_{St} \cdot v \text{ or}$$

$$P_{St} = \frac{F_{St} v}{3,600} = \frac{m g v \sin \alpha}{3600}$$

with P_{St} in kW, F_{St} in N, and v in km/h.
For a working approximation:

$$P_{St} = \frac{m g p v}{360,000}.$$

Refer to Table 6 for the necessary climbing power, normalized to 1000 kg vehicle mass, as a function of the speed v and the climbing resistance F_{St} (values are limited to a vehicle power-to-weight ratio of 72 kW/t).

The gradient is:

$$p = (h/l) \cdot 100 \% \text{ or}$$

$$p = (\tan \alpha) \cdot 100 \%,$$

with h as the height of the projected distance l .

In English-speaking countries calculations are made with the gradient.

Table 6: Climbing resistance and climbing power

Values at $m = 1000$ kg					
Climbing resistance F_{St} in N	Climbing power P_{St} in kW at various speeds				
	20 km/h	30 km/h	40 km/h	50 km/h	60 km/h
6500	36	54	72	—	—
6000	33	50	67	—	—
5500	31	46	61	—	—
5000	28	42	56	69	—
4500	25	37	50	62	—
4000	22	33	44	56	67
3500	19	29	39	49	58
3000	17	25	33	42	50
2500	14	21	28	35	42
2000	11	17	22	28	33
1500	8.3	12	17	21	25
1000	5.6	8.3	11	14	17
500	2.3	4.2	5.6	6.9	8.3
0	0	0	0	0	0

Example of calculating motive force and climbing power

To climb a hill with a gradient of $p = 21\%$, a vehicle weighing 1,500 kg will require approximately $1.5 \cdot 2000 \text{ N} = 3000 \text{ N}$ of motive force at the wheels (value from Table 5) and at $v = 40 \text{ km/h}$ roughly $1.5 \cdot 22 \text{ kW} = 33 \text{ kW}$ of climbing power (value from Table 6).

Motive force

The higher the engine torque M and overall transmission ratio i between the engine and drive wheels, and the lower the power-transmission losses, the higher the motive force F available at the driven wheels.

$$F = \frac{M}{r} \cdot i \cdot \eta \text{ or } F = \frac{P}{v} \cdot \eta.$$

η indicates the drivetrain efficiency. For a longitudinal engine $\eta \approx 0.88 - 0.92$, for a transverse engine $\eta \approx 0.91 - 0.95$.

The overall transmission ratio i for today's passenger cars is in the range of 1.5 to 20 and for off-road vehicles with additional reduction gears also much higher than this.

The motive force F is partially consumed in overcoming the running resistance F_w . Numerically higher transmission ratios are applied to deal with the substantially increased running resistance encountered on gradients (variable-speed transmission).

Vehicle and engine speeds

The following applies to the engine speed in min^{-1} (rpm) with the dynamic rolling radius r in m and v in m/s:

$$n = \frac{60 v i}{2 \pi r},$$

or with v in km/h:

$$n = \frac{1000 v i}{2 \pi 60 r}.$$

Likewise, the vehicle speed v can be calculated in m/s or in km/h from the drive speed according to the following formula:

$$v \left[\frac{\text{m}}{\text{s}} \right] = \frac{n \pi r}{30 i} \text{ or } v \left[\frac{\text{km}}{\text{h}} \right] = \frac{0.12 n \pi r}{i}.$$

Acceleration

The surplus force $F - F_W$ accelerates the vehicle. Or decelerates it when F_W exceeds F :

$$a = \frac{F - F_W}{k_m m} \quad \text{or} \quad a = \frac{P\eta - P_W}{v k_m m}.$$

During a translational acceleration of the vehicle the rotating parts of the drivetrain (wheels, flywheel, crankshaft etc.) are also rotationally accelerated. The additional rotational resistance force required for this purpose is taken into account in the form of the rotational inertia coefficient k_m as the apparent increase in vehicle mass due to the rotating masses. The rotational inertia coefficient is dependent on the ratio of the vehicle mass m to the mass moment of inertia of the drivetrain reduced to the powered axle. The latter is quadratically influenced by the overall transmission ratio i . The moment of inertia of the engine, which is approximately linear to the displacement V_H , has a significant influence (Figure 3).

Motive force and road speed on vehicles with automatic transmissions

When the formula for motive force is applied to automatic transmissions with hydrodynamic torque converters or hydrodynamic

clutches, the engine torque M is replaced by the torque at the converter turbine, while the rotational speed of the converter turbine is used in the formula for engine speed. The relationship between $M_{\text{Turb}} = f(n_{\text{Turb}})$ and the engine characteristic $M_{\text{Mot}} = f(n_{\text{Mot}})$ is determined using the characteristics of the hydrodynamic converter (see Hydrodynamic torque converter).

The tractive force (motive force) at the wheels for the individual gears as a function of the driving speed can be read from the tractive-force/running diagram (Figure 4). The kink typical of a hydrodynamic converter resulting from the end of torque multiplication can be seen in the first two gears. The converter is locked up by a clutch in higher gears. The top speed in each case as a function of gear and gradient can be determined from the points of intersection of the tractive-force lines with the total running-resistance lines (sum total of rolling resistance, aerodynamic drag, and climbing resistance). To the left of the point of intersection the motive force is greater than the running resistance and the vehicle is consequently accelerated. At the point of intersection the motive force is utilized entirely to overcome the running resistance and the vehicle speed thus remains constant.

Figure 3: Determining the rotational inertia coefficient k_m

V_H Displacement in liters,
 m Vehicle mass,
 i Overall ratio between engine and drive wheels,
 r Wheel radius.
 1 $m/V_H = 500 \text{ kg/l}$,
 2 $m/V_H = 750 \text{ kg/l}$,
 3 $m/V_H = 1000 \text{ kg/l}$.

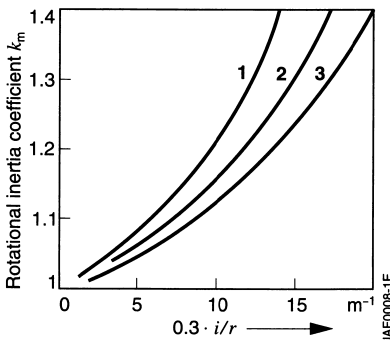
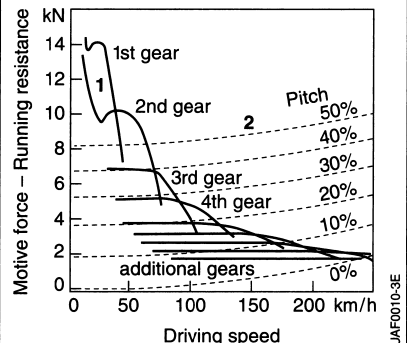


Figure 4: Tractive-force running diagram for a car with automatic transmission and hydrodynamic Trilok converter at full throttle

- 1 Running-resistance lines for different gradient,
- 2 Motive-force lines (tractive-force lines) for different gears.



Adhesion to road surface

Coefficient of static friction

The coefficient of static friction μ_r (between the tires and the road surface) is determined by the driving speed, the condition of the tires and the state of the road surface (Table 7). The figures cited apply to concrete and asphalt road surfaces in good condition. As a rule, they are in the region of $\mu < 1$. The coefficients of sliding friction (with wheel locked) are usually lower than the coefficients of static friction.

Tires for street-legal sports cars attain μ_r values of up to 1.3. Their rubber compounds are softer than in conventional tires and their treads are less pronounced.

Tires which are used in motor sport and in which special soft rubber compounds are used can provide friction coefficients of up to $\mu_r = 1.8$. These are for the most part smooth-tread slicks where the microgearing effects between rubber and road surface can be particularly pronounced by maximizing the tire contact patch.

Aquaplaning

Aquaplaning has a particularly dramatic influence on the contact between tire and road surface. It describes the state in which a layer of water separates the tire and the (wet) road surface (Figure 5). The phenomenon occurs when a wedge of water forces its way underneath the tire's

contact patch and lifts it from the road. The tendency to aquaplane is dependent upon such factors as the depth of the water on the road surface, the vehicle's speed, the tread pattern, the tread wear, and the load pressing the tire against the road surface. Wide tires are particularly susceptible to aquaplaning. An aquaplaning vehicle cannot transmit control and braking forces to the road surface, and may as result go into a skid.

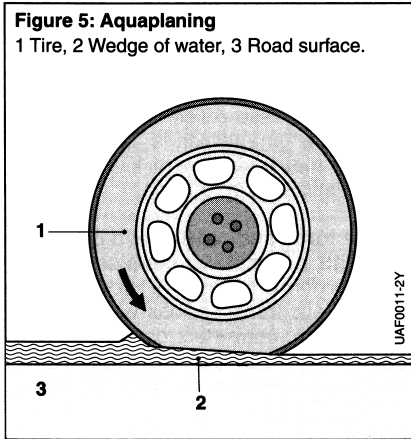


Table 7: Coefficients of static friction for pneumatic tires on various road surfaces

Driving speed in km/h	Tire condition	State of road surface				
		Dry	Wet, water level approx. 0.2 mm	Heavy rainfall, water level approx. 1 mm	Puddles, water level approx. 2 mm	Iced (black ice)
		Coefficient of static friction μ_r				
50	new	0.85	0.65	0.55	0.5	0.1 and less
	worn ¹	1	0.5	0.4	0.25	
90	new	0.8	0.6	0.3	0.05	—
	worn ¹	0.95	0.2	0.1	0.05	
130	new	0.75	0.55	0.2	0	—
	worn ¹	0.9	0.2	0.1	0	

¹ Worn to 1.6 mm tread depth (legal minimum in Germany as per § 36.2 StVZO (German Road Traffic Licensing Regulations)).

Accelerating and braking

The vehicle is regarded as accelerating or braking (decelerating) at a constant rate when a remains constant. The relations set out in Table 8 apply to acceleration from a standstill and braking (deceleration) to a standstill ($v = 0$).

Maxima for acceleration and braking (deceleration)

When the motive or braking forces exerted at the vehicle's wheels reach such a magnitude that the tires are just still within their

limit of adhesion (i.e., maximum adhesion is still present), the relationships shown in Tables 9 and 10 exist between the gradient angle α , coefficient of static friction μ_r , and maximum acceleration or deceleration. In this connection, k indicates the ratio between the load on driven or braked wheels and the total weight. If all wheels are driven or braked, $k = 1$. At 50 % load distribution $k=0.5$. In practice, the figures that can be achieved are always somewhat lower, as all the vehicle's tires do not simultaneously exploit their maximum possible adhesion during each accel-

Table 8: Accelerating and braking

	Equations for v in m/s	Equations for v in km/h
Acceleration or braking (deceleration) in m/s^2	$a = \frac{v^2}{2s} = \frac{v}{t} = \frac{2s}{t^2}$	$a = \frac{v^2}{25.92s} = \frac{v}{3.6t} = \frac{2s}{t^2}$
Acceleration or braking time in s	$t = \frac{v}{a} = \frac{2s}{v} = \sqrt{\frac{2s}{a}}$	$t = \frac{v}{3.6a} = \frac{7.2s}{v} = \sqrt{\frac{2s}{a}}$
Accelerating or braking distance in m	$s = \frac{v^2}{2a} = \frac{vt}{2} = \frac{at^2}{2}$	$s = \frac{v^2}{25.92a} = \frac{vt}{7.2} = \frac{at^2}{2}$

Table 9: Acceleration and braking (deceleration)

	Level road surface	Inclined road surface $\alpha; p = 100 \% \cdot \tan \alpha$	
Limit acceleration or deceleration $a_{\max}$ in m/s^2	$a_{\max} = k g (\mu_r - f_R)$	$a_{\max} = g (k (\mu_r - f_R) \cos \alpha \pm \sin \alpha)$ approximation ¹ : $a_{\max} \approx g (k (\mu_r - f_R) \pm 0.01p)$	+ when upgrade braking or downgrade acceleration – when upgrade acceleration or downgrade braking

Table 10: Achievable acceleration a_0 for given drive power P_A

a_0 in m/s , P_A in kW , v in km/h , m in kg .

Level road surface	Inclined road surface	
$a_0 = \frac{3600 P_A}{k m v}$	$a_0 = \frac{3600 P_A}{k m v} \pm g \sin \alpha$	+ Downgrade acceleration – Upgrade acceleration for $g \sin \alpha$ applies with approxi- mation ¹ $g p$ with p in %

Table 11: Work and power

	Level road surface	Inclined road surface $\alpha; p = 100 \% \cdot \tan \alpha$ [%]	
Drive or braking work W in J, see ²	$W = k m a s$	$W = m s (k a \pm g \sin \alpha)$ approximation ¹ : $W = m s (k a \pm g p/100)$	+ Downgrade braking or upgrade acceleration – Downgrade acceleration or upgrade braking
Drive or braking power in W at speed v	$P_A = k m a s v$	$P_A = m v (k a \pm g \sin \alpha)$ approximation ¹ : $P_A = m v (k a \pm g p/100)$	v in m/s . For v in km/h use $v/3.6$.

¹ Valid to approx. $p=20\%$ (under 2 % error), ² 1 J = 1 N m = 1 W s.

ation or deceleration. Electronic traction control and braking systems (TCS, ABS, ESC) maintain the traction level in the vicinity of the coefficient of static friction. It follows from the formula in Table 9 that on a level road surface with tires that exhibit $\mu_r < 1$ for all-wheel-driven vehicles (where $k=1$) the maximum acceleration is $g \cdot (1 - f_R)$.

Example:

$k = 0.5; g = 10 \text{ m/s}^2;$
 $\mu_r = 0.6; p = 15 \text{ \%};$
 $a_{\max} = 10 \cdot (0.5 \cdot 0.6 \pm 0.15) \text{ m/s}^2;$
upgrade braking (+): $a_{\max} = 4.5 \text{ m/s}^2,$
downgrade braking (-): $a_{\max} = 1.5 \text{ m/s}^2.$

Work and power

The power required to maintain a consistent rate of acceleration (deceleration) varies according to driving speed (Table 11). The power available for acceleration is:

$P_a = P \eta - P_W,$ where

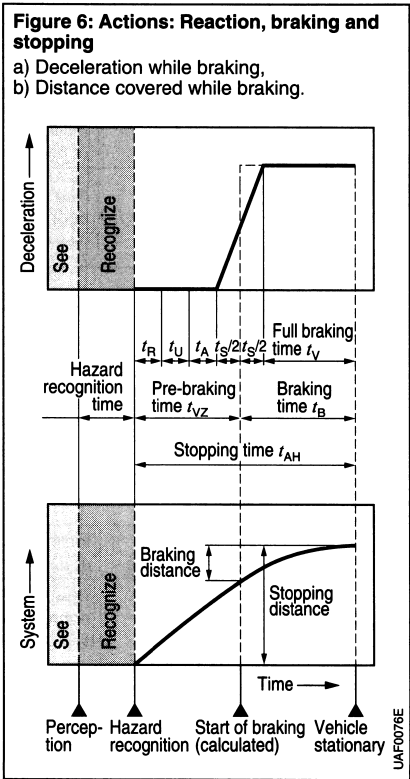
P Engine output,
 η Efficiency,
 P_W Motive power.

Actions: Reaction, braking and stopping

(in accordance with ÖNORM V 5050 [1] and [2])

Hazard recognition time

The hazard recognition time, also known as the danger reaction time, is the period of time that elapses between perceiving a visible obstacle or its movement and the time required to recognize it as a hazard (Figure 6). If, as part of this danger recognition and response process, it is necessary for the driver to turn his eyes towards the hazardous situation, the hazard recognition and danger reaction time will extend by roughly 0.4 s. It is further increased by poor concentration or tiredness.



Prebraking time

The prebraking time (t_{VZ}) is the period of time that elapses between the moment the hazard is recognized and the start of braking, defined by way of calculation. Based on the following formula, the prebraking time is in the range from approx. 0.8 to 1.0 s:

$$t_{VZ} = t_R + t_U + t_A + \frac{t_S}{2}.$$

The reaction time (t_R) is the period of time that elapses between the moment a defined incitement to action occurs and the start of the first specifically targeted action. Instinctive hazard recognition triggers an inherent, automatic reaction (spontaneous reaction, involuntary reflex), enabling the vehicle driver to determine both the point of the reaction as well as the position of the reason for the reaction, delayed by the distance covered during the prebraking time. Human beings require about 0.2 s for the spontaneous reaction; however, the reaction time will be at least 0.3 s if the driver needs to make a decision to perform a preventive or evasive action in response to conscious hazard recognition (choice reaction, voluntary action).

The transfer time (t_U) is the period of time the driver requires to transfer the foot from the accelerator pedal to the brake pedal. The transfer time is in the range of about 0.2 s.

The response time (t_A) is the period of time it takes to transmit the pressure applied at the brake pedal via the brake system through to the point when the braking action becomes effective (complete build-up of the application force and incipient increase in vehicle deceleration).

The pressure build-up time (t_S) is the period of time that elapses between the braking action taking effect and reaching fully effective braking deceleration. Alternatively, half the pressure build-up time ($t_S/2$) may be assumed as the start of braking, determined by way of calculation. In accordance with the EU Council of Ministers Directive EEC 71/320 Addendum 3/2.4 [3], the sum of the response time and pressure build-up time may not exceed 0.6 s. A poorly maintained braking system will lengthen the response and pressure build-up time.

Braking time

The braking time (t_B) is the period of time that elapses between the mathematically

Table 12: Stopping time and stopping distance

a in m/s^2 , v in m/s , t_{VZ} in s .

	Equations for v in m/s	Equations for v in km/h
Stopping time t_{AH} in s	$t_{AH} = t_{VZ} + \frac{v}{a}$	$t_{AH} = t_{VZ} + \frac{v}{3.6 a}$
Stopping distance s_{AH} in m	$s_{AH} = v t_{VZ} + \frac{v^2}{2 a}$	$s_{AH} = \frac{v}{3.6} t_{VZ} + \frac{v^2}{25.92 a}$

Table 13: Stopping distance as a function of driving speed and deceleration

Deceleration <i>a</i> in m/s²	Driving speed prior to braking in km/h													
	10	30	50	60	70	80	90	100	120	140	160	180	200	
	Distance during prebraking time (delay) of 1 s in m													
	2.8	8.3	14	17	19	22	25	28	33	39	44	50	56	
	Stopping distance in m													
4.4	3.7	16	36	48	62	78	96	115	160	210	270	335	405	
5	3.5	15	33	44	57	71	87	105	145	190	240	300	365	
5.8	3.4	14	30	40	52	65	79	94	130	170	215	265	320	
7	3.3	13	28	36	46	57	70	83	110	145	185	230	275	
8	3.3	13	26	34	43	53	64	76	105	135	170	205	250	
9	3.2	12	25	32	40	50	60	71	95	125	155	190	225	

calculated start of braking and the moment the vehicle comes to a complete stop. This period of time comprises half the pressure build-up time ($t_S/2$) (calculation assumption: only half the pressure build-up time but complete braking deceleration), as well as the full braking time (t_V), during which maximum braking deceleration is actually effective.

$$t_B = \frac{t_S}{2} + t_V.$$

Stopping time and stopping distance

The stopping time (t_{AH}) is the sum of the prebraking time (t_{VZ}) and braking time (t_B).

$$t_{AH} = t_{VZ} + t_B.$$

The stopping distance (s_{AH}) can be calculated by way of integration (Tables 12 and 13).

Safety margin

The minimum safety margin must be equal to the distance covered during the prebraking time t_{VZ} . The figure for a prebraking time of $t_{VZ} = 1.08$ s for speeds in km/h would be $0.3v$ meters; however, a minimum of $0.5v$ meters is advisable outside of built-up areas.

Passing (overtaking)

The complete passing maneuver involves pulling out of the lane, overtaking the other vehicle, and returning to the original lane (Figure 7). Passing can take place under a wide variety of very different circumstances and conditions, so precise calculations are difficult. For this reason, the following calculations, graphs, and illustrations will confine themselves to an examination of two extreme conditions: passing at a constant velocity and passing at a constant rate of acceleration. We can simplify graphic representation by treating the passing distance s_U as the sum of two (straight-ahead) components, while disregarding the extra travel involved by pulling out of the lane and back in again.

Passing distance

This passing distance is

$$s_U = s_H + s_L \text{ (variables in Table 14).}$$

The distance s_H which the more rapid vehicle must cover compared to the slower vehicle (considered as being stationary) is the sum of the vehicle lengths l_1 and l_2 and the safety margins s_1 and s_2 .

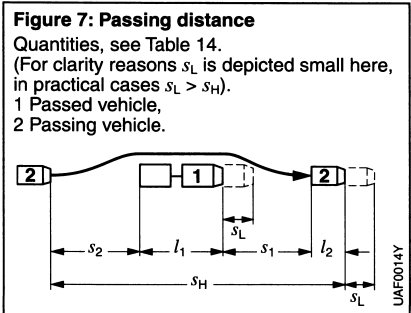
$$s_H = s_1 + s_2 + l_1 + l_2.$$

During the passing time t_U the slower vehicle covers the distance s_L which the faster vehicle has to travel in addition to s_H so that the safety margin is maintained (s_L in m, t_U in s):

$$s_L = \frac{t_U v_L}{3.6} \text{ (for } v \text{ in km/h).}$$

Table 14: Quantities and units

Quantity		Unit
a	Acceleration	m/s ²
l_1, l_2	Vehicle length	m
s_1, s_2	Safety margin	m
s_H	Relative distance traveled by passing vehicle	m
s_L	Distance traveled by vehicle being passed	m
s_U	Passing distance	m
t_U	Passing time	s
v_L	Speed of slower vehicle	km/h
v_H	Speed of faster vehicle	km/h



Passing at constant speed

On highways with more than two lanes, the overtaking vehicle will frequently be traveling at a speed adequate for passing before the actual process begins. The passing time (from initial lane change until return to the original lane has been completed) is then:

$$t_U = \frac{3.6 s_H}{v_H - v_L}$$

with t in s, s in m and v in km/h, and the passing distance is

$$s_U = \frac{t_U v_H}{3.6} \approx \frac{s_H v_H}{v_H - v_L}$$

Passing with constant acceleration

On narrow roads, the vehicle will usually have to slow down to the speed of the preceding car or truck before accelerating to pass. The attainable acceleration figures depend upon engine output, vehicle weight, speed, and running resistance. These generally lie within the range of 0.4 to 0.8 m/s², with up to 1.4 m/s² available in lower gears for further reductions in passing time. The distance required to com-

plete the passing maneuver should never exceed half the visible stretch of road.

Operating on the assumption that a constant rate of acceleration can be maintained for the duration of the passing maneuver, the passing time will be:

$$t_U = \sqrt{\frac{2s_H}{a}}$$

The distance which the slower vehicle covers within this period is defined as $s_L = t_U v_L / 3.6$; hence, the passing distance is:

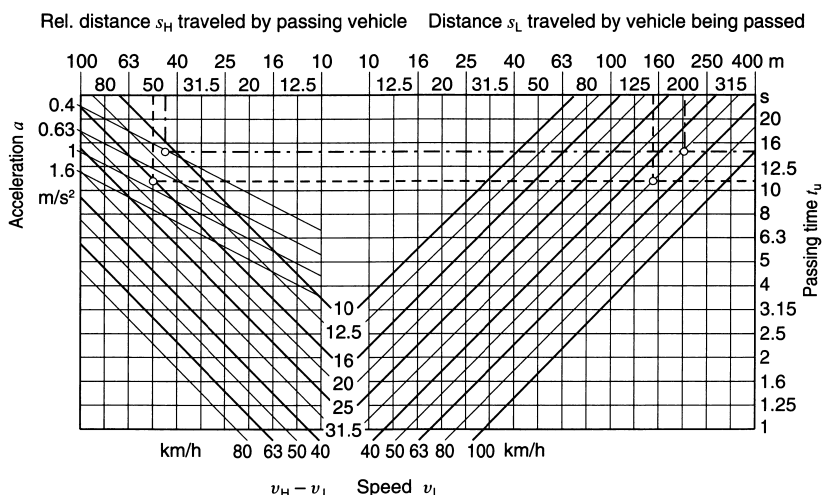
$$s_U = s_H + \frac{t_U v_L}{3.6}$$

with t in s, s in m and v in km/h.

The left side of Figure 8 shows the relative distances s_H for speed differentials $v_H - v_L$ and acceleration rates a , while the right side shows the distances s_L covered by the vehicle being passed at various speeds v_L . The passing distance s_U is the sum of s_H and s_L .

First, determine the distance s_H to be traveled by the passing vehicle. Enter this distance on the left side of the graph between the Y axis and the applicable

Figure 8: Graph for determining passing distance



($v_H - v_L$) line or acceleration line. Then extrapolate the line to the right, over to the speed line v_L .

Example 1 (constant acceleration):

$v_L = v_H = 50$ km/h (v at start of overtaking),
 $a = 0.4$ m/s²,
 $l_1 = 10$ m, $l_2 = 5$ m,
 $s_1 = s_2 = 0.3 |v_L| = 0.3 |v_H| = 15$ m,
 $\rightarrow s_H = 45$ m.

Solution

Enter intersection of $a = 0.4$ m/s² and

$s_H = 45$ m

in the left side of the graph.

Indication: $t_U = 15$ s, $s_L = 210$ m.

The following applies: $s_U = s_H + s_L = 255$ m.

Example 2 (constant speed):

$v_L - v_H = 16$ km/h,
 $v_H = 66$ km/h,
 $v_L = 50$ km/h,
 $s_1 = 0.3 |v_L| = 15$ m,
 $s_2 = 0.3 |v_H| = 20$ m,
 $\rightarrow s_H = 50$ m.

Solution

Intersection of $v_L - v_H = 16$ km/h with

$s_H = 50$ m

in the left side of the graph.

Indication: $t_U = 11$ s, $s_L = 150$ m.

The following applies: $s_U = s_H + s_L = 200$ m.

Visual range

For safe passing on narrow roads, the visibility must be at least the sum of the passing distance plus the distance which would be traveled by an oncoming vehicle while the passing maneuver is in progress. This distance is approximately 400 m if the vehicles approaching each other are traveling at speeds of 90 km/h, and the vehicle being overtaken at 60 km/h.

Fuel consumption

Specific fuel consumption

The most common way of presenting engine fuel-consumption maps is the "graph diagram", in which the effective mean pressure p_{me} and, as family parameters, lines of constant specific consumption (fuel throughput referred to power output in g/kWh) are plotted against the engine speed (Figure 9). This makes it possible to compare the efficiency of engines of different sizes and types.

Another way of presentation is, as a family parameter, the fuel throughput or mass flow (e.g., in kg/h). This form of presentation is particularly suitable as an input variable for CAE programs (computer-aided engineering), which can be used to simulate fuel consumption (corresponding to CO₂ emissions).

Entered in both presentations as additional information are the full-load torque curve as the upper limiting line, the idle and breakaway speeds as limiting speeds, and often also lines of constant power P (power hyperbolas on account of $P \sim p_{me} n$, with the engine speed n).

Consumption as sum of all losses and running resistances

If one disregards the influence of the driver on fuel consumption, which can run to 30 %, there are, based on the consumption formula (Figure 10), three distinct groups of influencing factors:

- the engine (including belt drive and ancillaries),
- the internal running resistances of the drivetrain (e.g., transmission, differentials), and
- the external running resistances.

External running resistances

The external running resistances determine a vehicle's minimum energy demand in a specified driving profile. They can be reduced by measures such as reduced vehicle mass, lower tire rolling resistance, and improved aerodynamics. On an average production vehicle without regenerative braking, 10 % reductions in mass, drag and rolling resistance result in fuel-consumption reductions of roughly 6 %, 3 %, and 2 %, respectively.

The formula in Figure 10 distinguishes between acceleration resistance and braking resistance. This illustrates clearly that consumption per unit of distance increases above all when the brakes are subsequently applied and not the overrun fuel cutoff of the engine or even a hybrid system is used to recover some of the brake energy.

Internal running resistances

The internal running resistances are made up of the losses in the drivetrain from the crankshaft to the wheels. In Figure 9, the sum total of the internal and external running resistances are shown by curves c and d, where vehicle A shows significantly lower resistances than vehicle B.

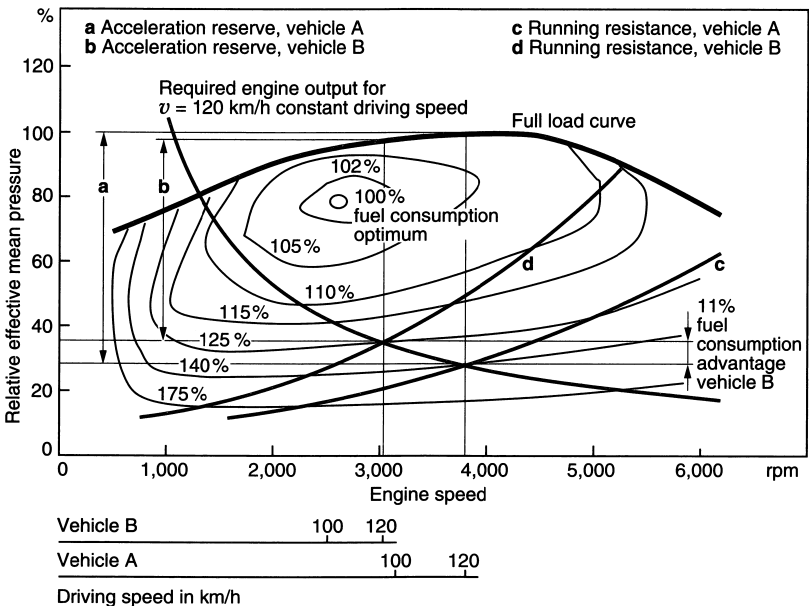
As well as the transmission losses in the drivetrain, the overall ratio also affects fuel consumption. This is calculated from the product of the transmission and differential ratios. The choice of overall ratio results in different operating points in the engine fuel-consumption map for

a specific driving speed. A "longer" ratio, i.e., smaller overall ratio, will generally move the operating points to map areas corresponding to lower fuel consumption. At the same time, it must be noted that the accelerating performance is reduced and the NVH behavior worsens ("Noise

Table 15: Quantities and units

Quantity	Unit
B_e Distance consumption	g/m
b_e Specific fuel consumption	g/kWh
m Vehicle mass	kg
A Frontal area of vehicle	m ²
f_R Coefficient of rolling resistance	—
c_W Drag coefficient	—
g Gravitational acceleration	m/s ²
t Time	s
v Driving speed	m/s
a Acceleration	m/s ²
B_r Braking resistance	N
η_0 Transmission efficiency of drivetrain	—
ρ Density of the air	kg/m ³
α Gradient angle	°

Figure 9: Engine fuel-consumption map (graph diagram)



Vibration Harshness", impacting on driving comfort). A useful choice of ratio is therefore only possible within limits.

Determining the distance-specific fuel consumption of conventional drives

The official data for standard fuel consumption are determined on the basis of dynamometer tests conducted on exhaust-emission test benches in which statutory test cycles (e.g., NEDC for Europe, FTP 75 and Highway for the USA and JC08 for Japan) are run. From 2017 and 2018 Europe and Japan will be switching to the WLTP cycle (Worldwide Harmonized Light-Duty Vehicles Test Procedure), which is intended to deliver more realistic consumption figures. The exhaust gas is collected in sample bags and its constituents HC, CO and CO₂ are subsequently analyzed for the purpose of determining consumption (see Exhaust-gas measuring techniques). The CO₂ content of the exhaust gas is proportional to the fuel consumption.

The following reference values apply to Europe:

Diesel: 1 l/100km ≈ 26.3 g CO₂/km.

Gasoline:

(Euro 4): 1 l/100km ≈ 24.0 g CO₂/km.

(Euro 5, E5): 1 l/100km ≈ 23.4 g CO₂/km.

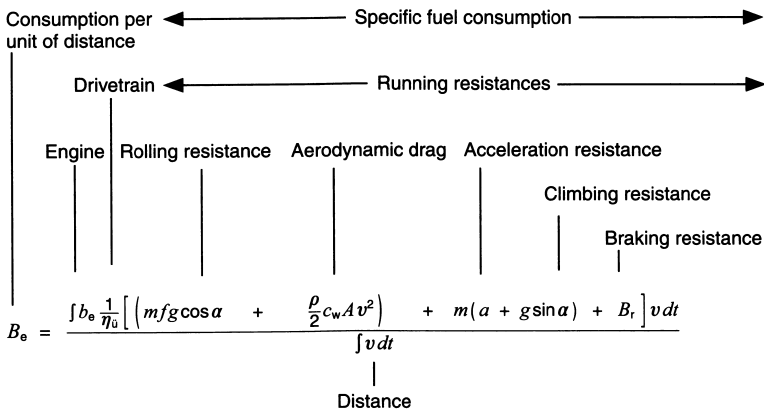
(Euro 6, E10): 1 l/100km ≈ 22.8 g CO₂/km.

The change from Euro 4 to Euro 5 and Euro 6 is based on a 5 % admixture of ethanol in gasoline (E5) and a 10 % admixture for Euro 6 (E10).

Vehicle masses are simulated on the test bench as an alternative by test masses. In today's certification cycles the test mass is, depending on the country and the curb weight of the vehicle, graded in increments of 55 to 120 kg. When the vehicle mass is allocated to the corresponding inertia weight class, the ready-for-operation mass of the vehicle (including all the fillers, tool kit and the fuel tank filled to 90 %) plus 100 kg as substitute for a driver and luggage is applied. The difference in consumption in the jump to an adjacent inertia weight class is – depending on the vehicle – between 0.15 and 0.25 l/100 km. In the WLTP cycle the actual vehicle mass is applied as the test mass.

Figure 10: Effect of vehicle design on fuel consumption

Quantities, see Table 15.



Units of fuel consumption

Standard fuel consumption is given in different units, depending on the country and the test cycle. In Europe, the figure is given in g CO₂/km or l/100 km, in the USA in mpg (miles per gallon), and in Japan in km/l.

Conversion examples:

30 mpg → 235.215/30 → 7.8 l/100 km,
22.2 km/l → 100/22.2 → 4.5 l/100 km.

Particularities of fuel consumption of autarkic hybrids

In the dynamometer testing of autarkic hybrids the battery current is determined as well as the CO₂ value and it is ensured in all the country-specific legislation that the battery state of charge is the same before and after the test. The permitted tolerance and possible corrections differ between Europe, USA, and Japan.

In hybrid vehicles some of the vehicle's kinetic energy is recovered and returned to the battery, where it is available for propulsion to assist the internal-combustion engine (boost mode) or for purely electric driving. The engine's operating point is moved by the electric motor into ranges in the engine map constituting lower fuel consumption (Figure 9) or unfavorable part-load ranges are avoided entirely. In this way, fuel savings of 10 % to 20 % compared with conventional vehicles can be achieved.

Particularities of fuel consumption of plug-in hybrids

Plug-in hybrids differ from autarkic hybrids in that the battery can be charged externally at a power socket or at a wall charging station (wallbox). Wall charging stations enable the battery to be charged at high power levels and thus reduce the charging time.

In contrast to autarkic hybrids and conventional vehicles, two dynamometer tests are performed on plug-in hybrids to determine consumption and the consumption results for fuel and electrical energy from the power socket are offset with different weighting depending on the legislation. Here a test is conducted with a fully charged battery and the cycle is repeated until the electric storage device

has reached its minimum charge. This is called the depleting test. A second test, the sustaining test, is performed at the minimum-charge level and is the same as the test for autarkic hybrids (Figure 12).

The utility factor (UF) weights the fuel consumption with a charged storage device (charge depleting test, CD) and a flat storage device (charge sustaining test, CS) as a function of the electric range. It can be determined from field data in accordance with SAE standard J2841 [4] or is defined by the legislature. Figure 11 shows a comparison of the cumulative utility factors for different countries (EU, USA, China, Japan) referred to the WLTP range. The higher the curve, the more the fuel consumption is weighted in the CD test compared with the CS test consumption for the same range, the utility factor increases and the certification consumption decreases. This is determined according to the following formula:

Certification consumption = sustaining consumption times (1 – utility factor).

Figure 11 shows that for an electric range in the WLTP cycle of 50 km the utility factor in China is 72 %, i.e., for sustaining consumption of 6 l/100 km the certification consumption would be 1.7 l/100 km.

The fuel consumption of a plug-in hybrid on the road can deviate markedly from the certification values since the share of electric driving depends, as well as on the driving style, very much on the customer's current-charging behavior. The charging behavior is decisively influenced by the existing charging infrastructure and the operating convenience.

Range and consumption of electric vehicles

Electric range

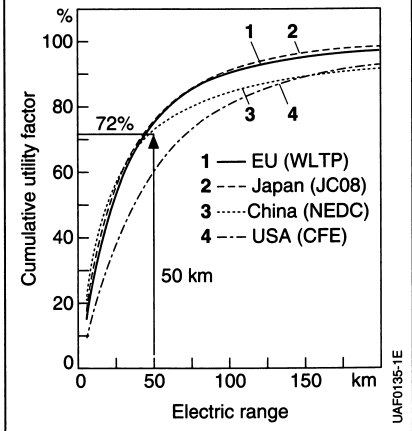
The official data for the standard range are determined on the basis of dynamometer tests on test benches in which statutory test cycles are repeatedly run. For example, for an electric range of 200 km the NEDC cycle must be run approximately 19 times. As in the dynamometer tests conducted on conventional vehicles and hybrids, secondary loads and air-conditioning influences are not taken into account here either. The latter play a large

role in electric vehicles with regard to the attainable range because all the energy for heating and cooling is drawn from the battery and thus is no longer available for propulsion. Figure 13 shows the air-conditioning influence for heating, cooling or dehumidifying (reheating), represented by the temperature, and the influence of driving behavior (speed, acceleration,

braking response), represented by different cycles, referred to the attainable range.

The higher the energy demand from the driving task, the less the effect of the A/C secondary loads. The smallest ranges are attained in urban cycles (using the example of the New York City Cycle, NYCC) at low temperatures. Constant-speed driving at temperatures around 20 °C provide for the largest ranges.

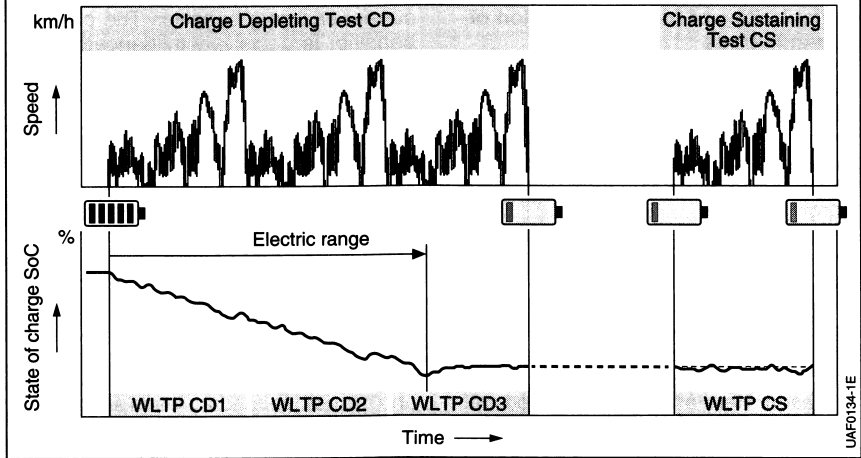
Figure 11: Utility factor as a function of the electric range in the WLTP cycle in different countries



Power-socket consumption

After the end of the range test the HV battery of each electric vehicle is recharged at a power socket or a wall charging station and the energy required for this is determined. Referring this energy to the previously determined range produces the power-socket consumption in Wh/km or kWh/100 km. While the terminal consumption, which can be measured at the battery terminal, is only dependent on the vehicle and the test cycle, the power-socket consumption additionally varies as a result of the charging procedure used. Figure 14 plots the dependence of the power-socket consumption on charging power and charging mode (charging procedure) for a concrete pair of vehicle and driving cycle. The different charging modes are alternating-current charging (AC) with Mode 2 and Mode 3

Figure 12: Plug-in test procedures, example WLTP



plugs and direct-current charging (DC). The higher the charging power, the lower the power-socket consumption and the higher the charging-system efficiency η . This is calculated as follows:

$$\eta = \frac{\text{power-socket consumption in } \frac{\text{kWh}}{100 \text{ km}}}{\text{terminal consumption in } \frac{\text{kWh}}{100 \text{ km}}}$$

The charging-system efficiency is affected by the losses from ancillary components (e.g., pumps, fans, and ECUs), the losses from the on-board charger during AC charging, and those from the wall charging station. The on-board charger here has the task of converting the alternating current from the charging network into direct current with the right voltage for the battery.

Driving performance of electric vehicles

The driving performance of electric vehicles is much more heavily dependent on temperature and the charge state of the energy accumulator than that of conventional vehicles. As the temperature decreases and with a low battery state of charge (SoC), the electric drivetrain delivers less power and consequently the top speed decreases and the acceleration time increases. At temperatures be-

low freezing and with the battery at 50 % charge the times double, the maximum speed decreases to approximately 50 % of the nominal top speed, and the acceleration time from 0 to 100 km/h doubles.

Figure 13: Relative range change as a function of temperature referred to the NEDC certification range

1 Constant 50 km/h, 2 Constant 100 km/h, 3 Constant 150 km/h, 4 Constant 200 km/h, 5 WLTP, 6 Long distance, 7 NYCC (urban).

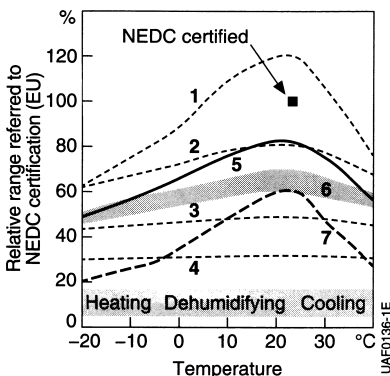
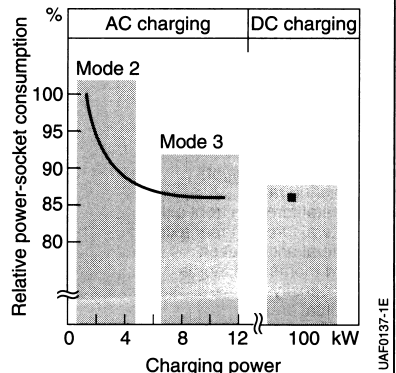


Figure 14: Electrical power-socket consumption as a function of charging power and charging procedure

Relative power-socket consumption is referred to the consumption at 1.4 kW charging power.



Dynamics of lateral motion

Ranges of lateral acceleration

Today passenger vehicles can reach lateral acceleration levels of up to 10 m/s². Lateral acceleration is subdivided into the following ranges (Figure 15):

The range from 0 to 0.5 m/s² is known as the small-signal range. The phenomenon to be considered in this range is the straight-running behavior, caused by crosswind and irregularities in the road, such as ruts. Crosswind excitation is induced by blustery, gusty wind and by driving into and out of wind shadow areas.

The range from 0.5 to 4 m/s² is known as the linear range, as the vehicle behavior that occurs in this range can be described with the aid of the linear single-track model. Typical maneuvers involving dynamics of lateral motion include sudden steering input, changing driving lanes as well as combinations of maneuvers involving dynamics of both lateral and longitudinal motion, such as load change reactions in turns.

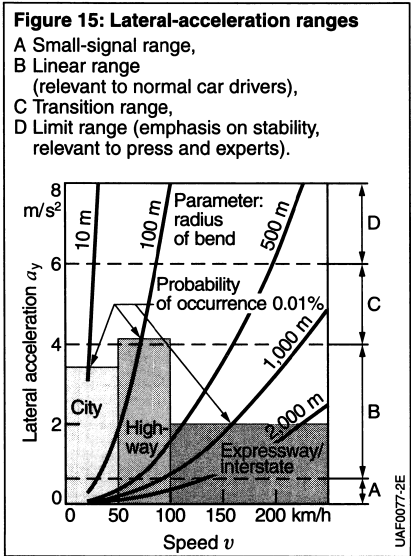
Table 16: Quantities and units

Quantity	Unit
δ	Axle steering angle
δ_H	Steering-wheel angle
α_v	Slip angle of front axle
α_h	Slip angle of rear axle
β	Float angle
ψ	Yaw angle
ω_o	Undamped natural frequency
l	Wheelbase
l_v	Distance between front axle and center of gravity
l_h	Distance between rear axle and center of gravity
v	Longitudinal velocity
v_r	Resulting wind impact velocity
C_v	Rear cornering stiffness
C_{h1}	Front axle
C_{h2}	Rear cornering stiffness
D	Rear axle
m	Damping factor
i_l	Total mass (weight)
F_{SV}	Steering ratio
F_{SH}	Lateral force on front axle
a_y	Lateral force on rear axle
θ	Lateral acceleration
ρ	Yaw moment of inertia
A	Air density
τ	Frontal area
F_S	Angle of impact
M_Z	Crosswind force
	Crosswind yaw moment

In the lateral acceleration range from 4 to 6 m/s², depending on their design features, passenger vehicles are categorized as either still linear or already nonlinear. This range is therefore considered to be a transition range. In this range, vehicles with maximum lateral acceleration of 6 to 7 m/s² (e.g. true offroad vehicles) already feature nonlinear characteristics, while vehicles that achieve higher levels of lateral acceleration (e.g. sports cars) still behave in line with linear characteristics.

The lateral acceleration range above 6 m/s² is reached only in extreme situations and is therefore referred to as the limit range. In this range, the vehicle characteristics are predominantly nonlinear, with the main emphasis shifted to vehicle stability. This range is reached on racing circuits or in situations leading up to accidents in normal road traffic.

The average driver generally drives in the range up to 4 m/s². This means both the small signal range as well as the linear range are relevant to the car driver when subjectively assessing the situation (Figure 15). For the average car driver, the probability of lateral acceleration occurring decreases exponentially with lateral acceleration.



speed. This draws the driver's attention to the increasing lateral acceleration.

The float-angle gradient (SG) can be calculated from Figure 16. The float angle gradient should be as low as possible in order to increase the stability of the vehicle [4, 5, 6].

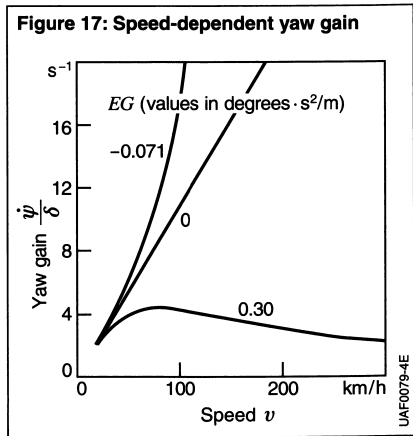
$$SG = \frac{d\beta}{da_y} = \frac{m}{C_h} \frac{l_v}{l}.$$

Yaw gain

The yaw gain defines the degree of yaw response that a vehicle executes in response to a steering angle in the quasi-steady-state range. The yaw gain factor can be determined by conducting the follow test procedure: When driving at a constant speed, the steering wheel is turned with sinusoidal motion at a frequency of less than 0.2 Hz. The steering-angle amplitude is selected in order to achieve a maximum lateral acceleration of about 3 m/s². Starting at a speed of 20 km/h, the maneuver is repeated at a speed that is increased by 10 km/h each time. Provided no aerodynamic influences occur at high speeds (lift or upthrust forces at the front and rear axles), the test will produce yaw gain curves that essentially agree with the following equation, derived from the linear single-track model [4, 5, 6]:

$$\left(\frac{\psi}{\delta}\right)_{\text{stat}} = \frac{v}{l + EG v^2}.$$

Figure 17 shows the yaw gain for a vehicle that tends to oversteer ($EG < 0$), that



has neutral steering ($EG=0$), and that understeers ($EG>0$). Only the vehicle that understeers is acceptable at high speeds and therefore has the right vehicle dynamics even when driving in a straight line. The speed at which a vehicle that tends to understeer has the greatest yaw response is known as the characteristic speed v_{char} . In the linear single-track model, this speed is expressed as:

$$v_{\text{char}} = \sqrt{\frac{l}{EG}}.$$

Damping factor

The following equilibrium of forces in the lateral direction is derived for the linear single-track model:

$$m a_y = F_{sv} \cos \delta + F_{sh}.$$

For the torque balance:

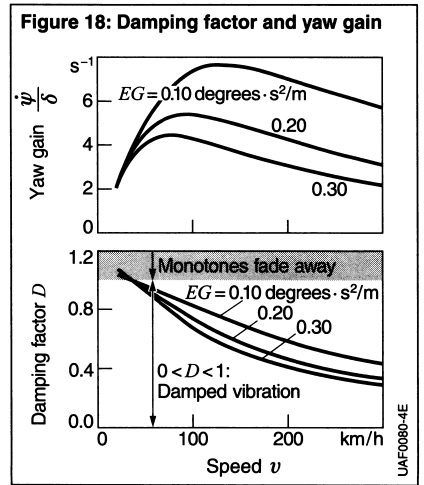
$$\theta \ddot{\psi} = F_{sv} l_v \cos \delta + F_{sh} l_h.$$

The damping factor D for excitation in terms of the dynamics of linear motion can be derived from the two equations:

$$D = \frac{1}{\omega_e} \left(\frac{C_v + C_h}{m v} + \frac{C_v l_v^2 + C_h l_h^2}{\theta v} \right).$$

The following equation expresses the undamped natural frequency:

$$\omega_e = \sqrt{\left(\frac{C_h l_h - C_v l_v}{\theta} + \frac{C_v C_h l^2}{\theta m v^2} \right)}.$$



The damping factor of a vehicle can be identified, for example, from the yaw response of sudden steering or step input. The vehicle is designed to ensure that damping is as high as possible.

Figure 18 shows the damping factor and yaw gain for various self-steering gradients. This results in the following conflict of objectives:

- A high self-steering gradient is required if the vehicle is to have good straight-running characteristics.
- The self-steering gradient must be as low as possible to facilitate a high damping factor, particularly at high speeds.

Lateral-agility diagram

A further important variable governing vehicle balance is the total steering ratio i_t . The steering-wheel angle, together with the total steering ratio i_t , is calculated from the axle steering angle:

$$\delta_H = i_t \delta.$$

This results in the following equation for maximum yaw gain:

$$\left(\frac{\dot{\psi}}{\delta_H}\right)_{\max} = \frac{1}{2i_t \sqrt{l EG}}.$$

This maximum is plotted in the lateral-agility diagram (Figure 19) as a function of the steering ratio. The diagram additionally shows the EG isolines. The self-steering gradient is constant along these curves. The desired target ranges for the yaw gain and steering ratio can be plotted in this diagram for the purpose of defining the necessary self-steering gradients.

If only the steering ratio is changed in a vehicle, the maximum yaw gain can be determined in the lateral-agility diagram by shifting the baseline along the EG isolines. The shift will be along the vertical axis if the axis characteristics are varied.

Dynamics of lateral motion caused by crosswind

Wind can induce lateral dynamics in motor vehicles. The vehicle responds to this external influence by drifting off course, lateral acceleration, and a change in yaw angle and roll angle. The driver then attempts to take corrective action to counteract this discrepancy. Consequently, the response capabilities of the

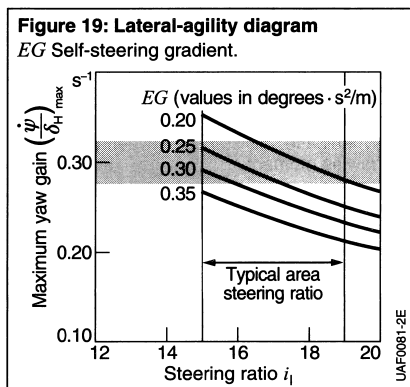
driver as well as the correctability of the vehicle are taken into consideration in a second stage. According to current findings, the direct response of the vehicle to crosswind is the principal variable for subjectively assessing overall vehicle stability in crosswinds. This offers the advantage that the interaction between crosswind and vehicle response can be effectively observed by analysis.

Characteristically, the average car driver perceives two states induced by wind excitation:

- Natural crosswind, which can vary in terms of direction and wind velocity while driving.
- Driving into and out of wind shadow areas where forces of greatly varying strength can act on the vehicle.

The automotive industry strives to minimize the effects of excitations triggered by wind forces by taking the following vehicle factors into consideration:

- “Cornering stiffness” of the tires, i.e. to what extent the lateral force changes as the slip angle increases. The wheel load of the tire remains constant in this consideration.
- Total weight rating of the vehicle.
- Position of center of gravity.
- Axle characteristics.
- Equilateral and reciprocal suspension.
- Damping.
- Kinematics and elastokinematics of the axles.
- Aerodynamic shape and frontal area of the vehicle.



Aerodynamic forces and moments

When a vehicle moves at velocity v in a wind at a speed v_w , the vehicle will be subject to wind impact applied at the resulting velocity v_r . In connection with natural crosswind, the angle of impact τ generally differs from 0 degrees and therefore generates a lateral force F_s and a yawing moment M_z that acts on the vehicle.

In aerodynamics, it is standard practice to specify dimensionless coefficients instead of forces and moments. Therefore:

$$F_s = c_s \frac{\rho}{2} v_r^2 A \ ; \ M_z = c_M \frac{\rho}{2} v_r^2 A l .$$

The moment M_z and the lateral force F_s , which is defined at the mid-point of the wheelbase, can be represented by a single lateral force F_s when the point of impact is located at the center of the pressure point D (Figure 20). The distance d between the aerodynamic reference point B and the center of the pressure point D is calculated as follows:

$$d = \frac{M_z}{F_s} = \frac{c_M}{c_s} \frac{l}{c_s} .$$

To keep aerodynamic influences as low as possible, appropriate steps should be taken to ensure that the center of the pressure point D is as close to the vehicle's center of gravity S as possible. This consequently reduces the effective impact of the moment.

Figure 21 represents the aerodynamic coefficients of the two most typical structural shapes of vehicles, i.e. the station

wagon and sedan, as a function of the angle of impact τ . The resulting distance d is considerably lower for stations wagons than for sedans (Figure 20). For vehicles with the center of gravity located at the mid-point of the wheelbase, the station wagon structure is therefore less sensitive to crosswind than the sedan structure.

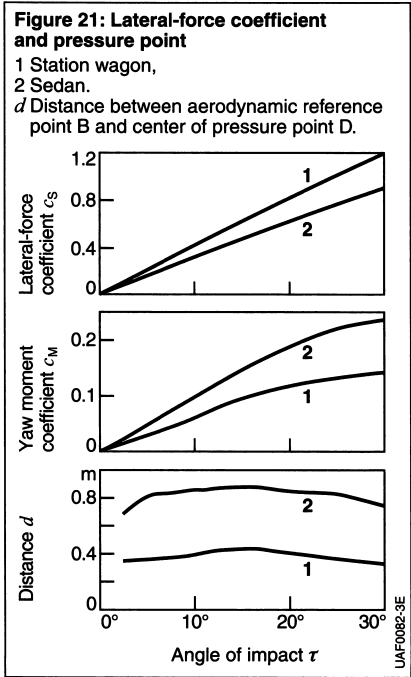
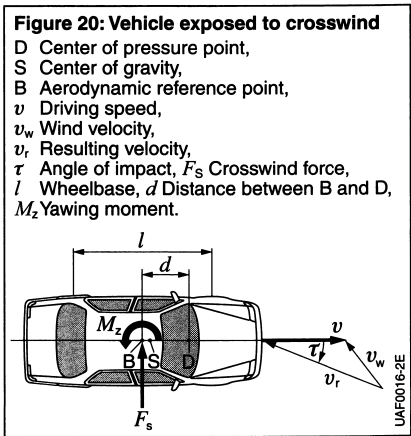
Cornering behavior

Centrifugal force in turns

$F_{ct} = \frac{m}{r_k} \frac{v^2}{r_k}$ (see Figure 22).

Body roll in turns

In turns, the centrifugal force which concentrates around the center of gravity causes the vehicle to tilt away from the path of travel. The magnitude of this rolling motion depends upon the rates of the springs and their response to alternating compression, and upon the lever arm of the centrifugal force (distance between the roll axis and the center of gravity). The roll axis is the body's instantaneous axis of rotation relative to the road surface.



Like all rigid bodies, the vehicle body consistently executes a screwing or rotating motion; this motion is supplemented by a lateral displacement along the instantaneous axis.

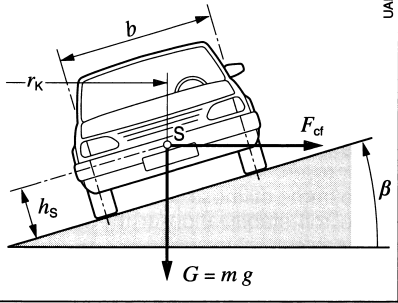
The higher the roll axis, i.e. the closer it is to a parallel axis through the center of gravity, the greater will be the transverse stability and the less the roll in turns. However, this generally implies a corresponding upward displacement of the instantaneous axes of the wheels, resulting in a change in track width (with negative effects on driving safety). For this reason, designs seek to combine a high instantaneous roll center with minimal track change. The goal therefore is to place the instantaneous axes of the wheels as high as possible relative to the body, while simultaneously keeping them as far from the body as possible.

A frequently applied means for finding the approximate roll axis is based on determining the centers of rotation of an equivalent body motion. This body motion takes place in those two planes through the front and rear axles which are vertical relative to the road. The centers of rotation are those (hypothetical) points in the body which remain stationary during the rotation. The roll axis, in turn, is the line which connects these centers (instantaneous roll centers). Graphic portrayals of instantaneous roll centers are based on a rule according to which the instantaneous poles of rotation of three systems in a state of relative motion lie along a common pole line.

The complexity of the operations required for a more precise definition of the spatial relationships involved in wheel motion make it advisable to employ general, three-dimensional simulation models [4, 5, 6].

Figure 22: Centrifugal force in turns

- b Lane width,
- h_s Height of center of gravity,
- r_k Turn/curve radius,
- β turn/curve camber,
- G Vehicle weight,
- S Center of gravity,
- F_{cf} Centrifugal force.



References for dynamics of lateral motion

- [1] ÖNORM V 5050: Straßenverkehrsunfall und Fahrzeugschaden; Terminologie.
- [2] Fritz Sachser in Fucik, Hartl, Schlosser, Wielke (Editors): Handbuch des Verkehrsunfalls, Part 2, Manz-Verlag Vienna, 2008.
- [3] Council Directive of 26 July 1971 on the approximation of the laws of the Member States relating to the braking devices of certain categories of motor vehicles and their trailers.
- [4] B. Heissing, M. Ersoy: Fahrwerkhandbuch, 2nd Edition, Vieweg + Teubner Verlag, 2008.
- [5] H.-P. Willumeit: Modelle und Modellierungsverfahren in der Fahrzeugdynamik, Teubner-Verlag, 1998.
- [6] M. Mitschke, H. Wallentowitz: Dynamik der Kraftfahrzeuge, 4th Edition, Springer-Verlag 2004.

Table 17: Critical speeds in turns/curves (numerical-value equations)

	Flat curve	Banked curve
Speed at which the vehicle exceeds the limit of adhesion (skid)	$v \leq 11.28 \sqrt{\mu_r r_k} \text{ km/h}$	$v \leq 11.28 \sqrt{\frac{(\mu_r + \tan \beta) r_k}{1 - \mu_r \tan \beta}} \text{ km/h}$
Speed at which the vehicle tips	$v \geq 11.28 \sqrt{\frac{b r_k}{2 h_s}} \text{ km/h}$	$v \geq 11.28 \sqrt{\frac{\left(\frac{b}{2 h_s} + \tan \beta\right) r_k}{1 - \frac{b}{2 h_s} \tan \beta}} \text{ km/h}$

h_s Height of center of gravity (in m), μ_r Max. coefficient of friction, b Lane width (in m), r_k Turn/curve radius (in m), β Turn/curve camber

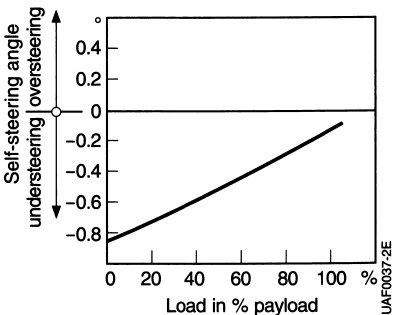
Special operating dynamics for commercial vehicles

Self-steering effect

The aim is to achieve non-problematic, understeering handling. Stationary vehicle handling is defined by the self-steering effect. The self-steering effect is determined by:

- The mechanical and hydraulic parameters of the steering gear.
- The stiffness and geometry of the steering linkage.
- The elastokinematics of the front and rear axles.
- The frame stiffness.
- The roll stabilization at the front and rear axles.

Figure 1: Self-steering angle of an 18-ton truck cornering at 3 m/s² of lateral acceleration



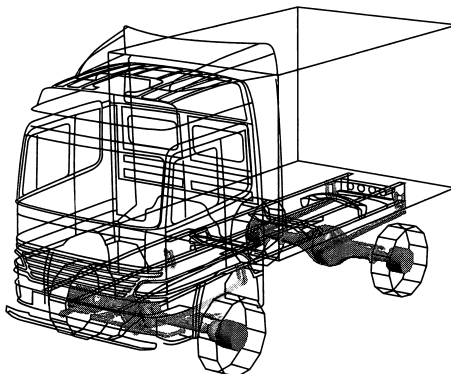
External interference factors such as road-surface irregularities or crosswind, for example, must not give rise to any great disruptions of vehicle motion. The elastokinematic design of the axles serves to minimize these interference factors. The aim is to minimize steering movements as a result of parallel and alternating compression and rebound and as a result of braking. The kinematic points of the steering or the connecting points of the leaf springs, for example, act as the setting levers in the configuration of the elastokinematics. Geometrically non-linear finite-element programs are used to configure the elastokinematics. The configuration is monitored on elastokinematic test benches.

Heavy-duty trucks with air suspension are normally equipped with solid or rigid axles controlled with suspension arms and links. Such axles are generally designed to ensure that the proportion of self-steering effects in axle control is virtually constant for all laden states since there is no difference in level between the “unladen” and “laden” states. To date, axle control concepts involving independent wheel control have only been implemented on light utility vans and buses.

Wheel loads at the truck’s rear axle vary dramatically, depending on whether the truck is “unladen” or “laden”. This leads to the vehicle responding to reductions in load with more pronounced understeer (Figure 1).

Figure 2: Multibody-system model of an all-wheel drive vehicle

for determining self-steering effects and simulating operating dynamics.



In vehicles with multiple non-steered rear axles, such as 6×4 or 6×2 (six wheels, four or two of which are driven), a constraining torque is generated around the vehicle's vertical axis on account of the different slip angle at the first and second rear axles. The lateral force on the axles that results from this is relevant for the maneuvering system and can be determined as follows (Figure 3).

Cornering forces resulting from constraint for small angles α (For sizes, see Figure 3):

$$F_{S1} = F_{S2} - F_{S3}, \text{ where}$$

$$F_{S2} = c_{p2} n_2 \alpha_2,$$

$$F_{S3} = c_{p3} n_3 \alpha_3.$$

Slip angle:

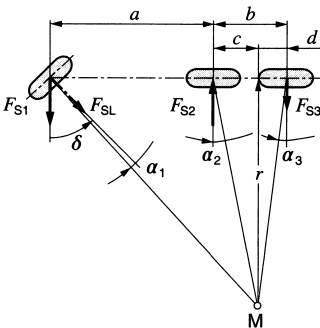
$$\alpha_2 = \frac{1}{r} \cdot \frac{c_{p3} n_3 b (a+b)}{c_{p3} n_3 (a+b) + c_{p2} n_2 a},$$

$$\alpha_3 = \frac{b}{r} - \alpha_2.$$

α_i is in the radian, c_{p2} and c_{p3} are the slip stiffnesses from the tire maps, n_2 and n_3 the number of tires per axle.

Figure 3: Cornering forces F_{Si} and slip angle α on a three-axle vehicle with non-steered tandem axle

- a Distance, front axle to first rear axle,
- b Distance, first rear axle to second rear axle,
- c Distance, first rear axle to instantaneous center,
- d Distance, second rear axle to instantaneous center,
- F_{Si} Cornering forces, $i = 1, 2, 3, L$,
- α_i Slip angle, $i = 1, 2, 3$,
- δ Steering angle,
- r Radius of curve,
- M Instantaneous center.



UAF0038-2Y

Tipping resistance

As the vehicle's total height increases, the vehicle will also have an increasing tendency to tip to the side in a turn before it starts to slide. In the same way as the overall dynamic behavior, the tipping limits of vehicles are determined by means of multibody-system simulations (multibody system, Figure 2). The simulations investigate various stationary and non-stationary maneuvers, e.g. steady-state circular-course driving ([1]) and double lane change [2]. The achievable lateral acceleration b at the tipping limit during steady-state circular-course driving is $b = 6$ to 8 m/s^2 for light utility vans, $b = 4$ to 6 m/s^2 for trucks and $b \approx 3 \text{ m/s}^2$ for double-decker buses.

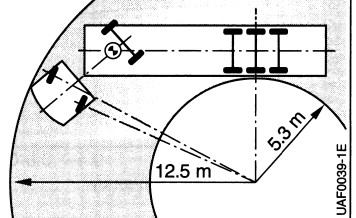
The increasing use of driving-dynamics control systems (Electronic Stability Program) also on commercial vehicles can largely reduce the risk of tipping by estimating or determining the load status and load distribution.

Width requirement

The width requirement of motor vehicles and semi-trailer combinations is greater during cornering than when the vehicle moves in a straight line. With respect to selected driving maneuvers, it is necessary to determine the radius described by the vehicle's outer extremities during cornering, both in order to ascertain its suitability for certain applications (e.g. narrow transit routes through constricted areas) and to confirm compliance with legal regulations (Figure 4). This is determined using multibody-system simulations.

Figure 4: Turning-circle regulation [8] on example of a semi-trailer unit

Representation of the permissible steady-state cornering radii.



UAF0038-1E

Handling characteristics

Objective analyses of vehicle handling are based on various driving maneuvers such as steady-state circular-course driving [1], single and double lane change [2], step input [3], weave test [4], sinusoidal steering input and frequency response [5] and braking in a straight line on μ split [6] and braking during a turn/curve [7].

The dynamic lateral response of truck-trailer combinations (e.g. semitrailer, truck-trailer) generally differs from that of rigid vehicles. Particularly significant are the distribution of loads between truck and trailer or semitrailer, and the design and geometry of the mechanical coupling device within a given combination.

Impairments of straight-running stability due to vehicle yaw are induced by rapid steering corrections associated with evasive maneuvers, gusts of crosswind, uneven road surfaces, obstacles on one side, ruts in the road surface, and cross-fall cambers. The yaw oscillations caused by these impairments must decay rapidly if vehicle stability is to be maintained. These yaw oscillations can be assessed on the basis of the yaw-velocity frequency response (Figure 5). The yaw-velocity frequency

responses of different truck-trailer combinations demonstrate excessive increase in resonance in the least favorable case (tractor-trailer unladen, center-axle trailer laden, curve 2 in Figure 5). This type of combination demands a high degree of driver skill and a defensive driving style.

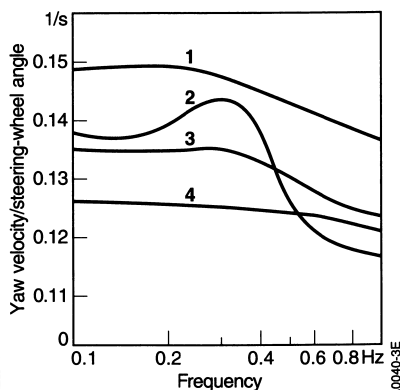
With semitrailer units, braking maneuvers undertaken under extreme conditions can induce jackknifing. This process is initiated when, on a slippery road surface (μ -low), loss of lateral force is induced by excess braking force applied to the tractor's rear axle or due to excess yaw moment under μ -split conditions. Installation of an driving-dynamics control system (Electronic Stability Program) represents the most effective means of preventing jackknifing.

References special operating dynamics for commercial vehicles

- [1] ISO 14792 (2011): Road vehicles – Heavy commercial vehicles and buses – Steady-state circular tests.
- [2] ISO 3888-1 (1999), ISO 3888-2 (2011): Passenger cars – Test track for a severe lane-change maneuver – Part 1: Double-lane change Part 2: Obstacle avoidance
- [3] ISO 14793 (2011): Road vehicles – Heavy commercial vehicles and buses – Lateral transient response test methods.
- [4] ISO 11012 (2009): Heavy commercial vehicles and buses – Open-loop test methods for the quantification of on-center handling – Weave-test and transition test.
- [5] ISO 7401 (2011): Road vehicles – Lateral transient response test methods – Open-loop test methods.
- [6] ISO 16234 (2006): Heavy commercial vehicles and buses – Straight-ahead braking on surfaces with split coefficient of friction – Open-loop test method.
- [7] ISO 14794 (2011): Heavy commercial vehicles and buses – Braking in a turn – Open-loop test methods.
- [8] Commission Directive 2003/19/EC of 21 March 2003 for changing the Guideline 97/27/EC of the European Parliament and of the Council about the weights and dimensions of certain classes of motor vehicles and motor-vehicle trailers regarding the adaptation to the technical progress.

Figure 5: Yaw-velocity frequency responses

- 1 Semi-trailer unit (laden),
- 2 Truck-trailer unit (unladen with laden center-axle trailer),
- 3 Truck-trailer unit (laden),
- 4 Truck (laden).



UFA040-3E

Driving-dynamics test procedures as per ISO

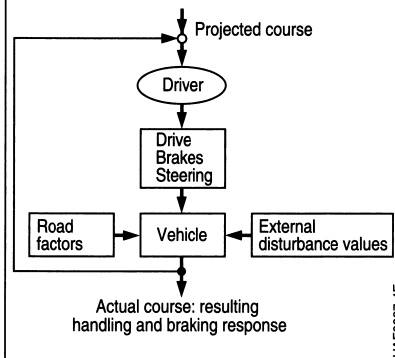
The science devoted to studying the dynamics of vehicle handling generally defines its subject as the overall behavior of the entire system represented by “driver – vehicle – environment” (Figure 2). The driver assesses the vehicle’s handling qualities based on the sum total of his/her subjective impressions. Internationally standardized test procedures have been developed since the end of the 1970s. On the one hand, these test procedures are intended to describe vehicle handling as objectively and consistently as possible. On the other hand, they serve to determine characteristic variables which correlate well with the subjective driving impressions. Furthermore, it is possible to use the standardized test procedures to compare test results with simulation results under identical boundary conditions.

Most of the internationally standardized procedures are carried out in an open loop, i.e., with a defined steering input without controlling intervention by the driver. In this way, vehicle handling is determined without the influence of different steering strategies of individual drivers. Only a few procedures are carried out in a closed loop with the input of a specific course which can be taken by the driver in an individual way.

There are currently a total of roughly 20 ISO standards for driving-dynamics procedures, of which the following maneuvers used most frequently in vehicle development will be described here:

- Steady-state circular-course driving [1], [2].
- Test procedures for transient response [3], [4].
- Weave test and transition test [5], [6].
- Braking in a turn/curve [7], [8].
- Closing curve [19].

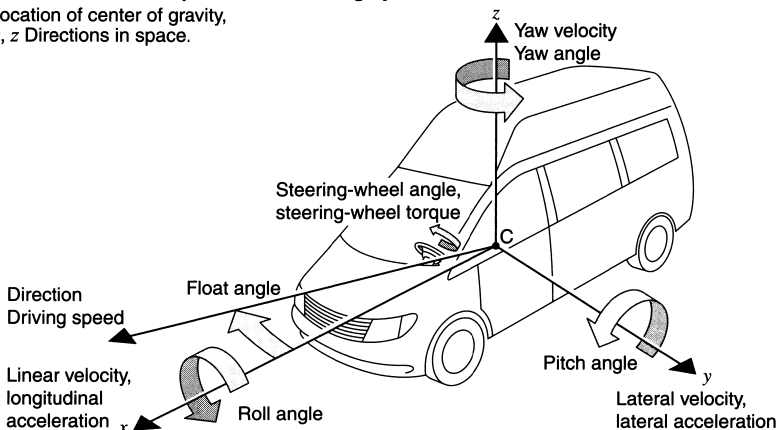
Figure 2: Overall system “Driver – Vehicle – Environment” as closed loop



LUA0027-4E

Figure 1: Assessment quantities for driving dynamics

S Location of center of gravity,
x, y, z Directions in space.



SAF0028-2E

Further test procedures not explicitly described here are, for example:

- Double lane change (closed loop, [11]).
- Lateral stability of vehicle combinations [12], [13].
- Crosswind stability [14].
- Load changes when cornering [15].

These test procedures were originally created for automobiles ([9], [10]). Test methods based on these standards were later developed for heavy-duty commercial vehicles in order to cater for the unique handling characteristics of heavy-duty commercial vehicles with their large mass and inertia figures.

General boundary conditions which must be equally observed for all driving-dynamics test procedures, such as, for example, the condition of the road surface, environmental conditions and tires, are defined in a separate ISO standard [16], which

additionally defines what is required of the measuring techniques for driving-dynamics measurements.

Assessment quantities

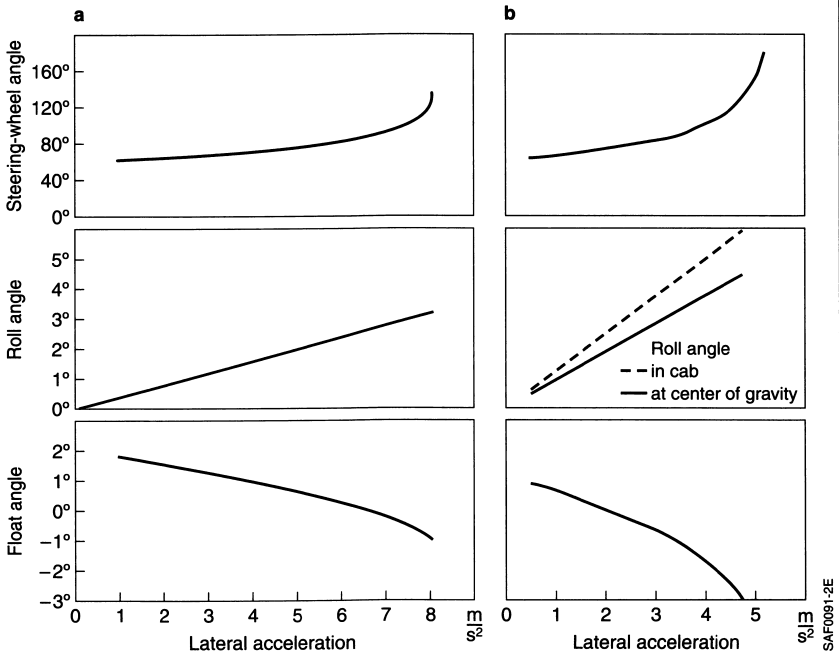
The following measurable quantities are primarily used to assess vehicle handling (Figure 1):

- Steering-wheel angle and steering-wheel torque.
- Lateral acceleration.
- Yaw velocity.
- Roll angle.
- Float angle.

Depending on the test procedure defined, characteristic values are determined from these and other quantities for the purpose of describing and assessing vehicle handling. These measured values and characteristic quantities are defined in a separate ISO standard [17].

Figure 3: Steady-state circular-course driving

- a) Passenger car, 40 m circle radius,
b) 40-ton semitrailer unit, loaded, 80 m circle radius.



Steady-state circular-course driving Performance

The "Steady-state circular-course driving" test procedure ([1], [2]) is generally carried out in such a way that the vehicle is accelerated on a circular course with a constant radius (a radius of 40 m is usually selected for passenger cars, 80 m for trucks) starting from a very slow starting speed up to the maximum attainable vehicle lateral acceleration. The longitudinal acceleration here should not exceed approximately 1 m/s^2 so that the driving state can be viewed as sufficiently steady-state. When semitrailer-truck or truck-trailer combinations are measured, the measurement is conducted at a number of constant driving speeds on a circular-course radius.

Assessment

The assessment criteria primarily used are the curves of steering-wheel angle, roll angle and float angle plotted against lateral acceleration, which are shown in Figure 3 for a passenger car and for a truck with semitrailer.

From the curve of steering-wheel angle against lateral acceleration it is possible to determine the vehicle's self-steering effect using a single-lane model (see Dynamics of lateral motion). Today's vehicles, both passenger cars and heavy-duty commercial vehicles, are basically designed as understeering. In other words, as their speed increases on a constant circular-course radius, they require a significant increase in steering angle. The limit range of an understeering vehicle is determined by the maximum transferable cornering force of the front wheels, which is recognizable from the sharply progressively rising steering-wheel angle with high lateral acceleration.

The curve of roll angle against lateral acceleration describes the lateral inclination of the vehicle that can be perceived by the driver. This is greatly dependent on the vehicle load, particularly in the case of commercial vehicles. The upshot of this may be that, in the case of commercial vehicles with a high loading center of gravity, the maximum achievable lateral acceleration is determined not by the cornering force of the tires, but by the tipping limit of the vehicle.

In the case of trucks, the roll angle is additionally determined not only at the center of vehicle gravity, but also, due to the torsionally weak frame and the separate driver's cab mounting, at several fixed measuring points.

The curve of float angle against lateral acceleration is an indicator of the lateral stability of the vehicle that can be perceived by the driver and is determined above all by the properties of the tires. In addition to the tire properties, the following vehicle parameters have a significant influence on stationary vehicle handling:

- Load and axle-load distribution.
- Stiffness of springs and stabilizers.
- Kinematic and elastokinematics movements of the wheel suspension.

Transient response

The test procedures for transient response ([3], [4]) serve to determine the vehicle response to rapid, dynamic steering stimulations, e.g., during rapid evasion maneuvers. Procedures frequently used in tests and simulations are "Step input" and "Sinusoidal steering input and frequency response".

Step input

The "Step input" test procedure involves the vehicle, starting from straight-ahead driving at a constant speed, being turned in with a very fast steering movement from the zero steering-wheel position to a stationary steering-wheel angle, resulting in circular driving with a defined, constant lateral acceleration. An arrangement usual for passenger cars involves turning in with a steering-wheel angle velocity of approximately $360^\circ/\text{s}$ to a lateral acceleration of 4 m/s^2 at a driving speed of 80 km/h .

The time delay and the overswing values with which the quantities of yaw velocity, lateral acceleration, roll angle, and float angle respond to the steering-wheel angle input are determined during this maneuver (Figure 4). The vehicle should, on the one hand, not respond too sluggishly to the steering movement and, on the other hand, not respond with excessive overswing to the steering input.

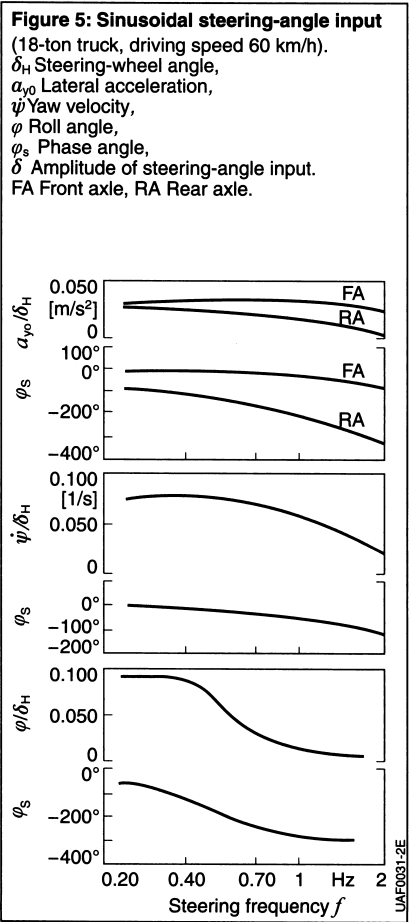
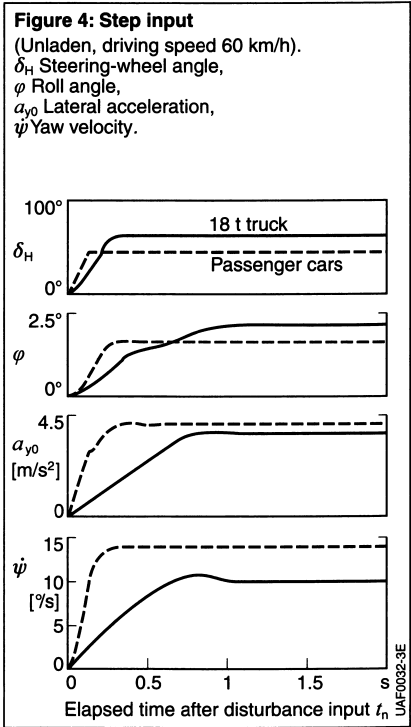
Sinusoidal steering input and frequency response

The “Sinusoidal steering response” test procedure involves the vehicle being excited at a constant driving speed with a sinusoidal steering-wheel angle signal whose amplitude is kept constant and whose frequency is increased starting at slow steering movements of 0.2 Hz to fast steering movements up to 2.0 Hz. The steering amplitude is as a rule selected in such a way that the vehicle remains in the linear driving range, for example in the case of passenger cars usually in such a way that a lateral acceleration of max. 4 m/s² is obtained at the slowest steering frequency and a driving speed of 80 km/h. This makes it possible to assess the dynamic vehicle handling in the entire steering-frequency range to be excited by the driver.

For assessment, the amplitudes referred to the input variable of steering-wheel angle and phase angles of the

quantities of steering-wheel torque, yaw velocity, lateral acceleration, roll angle, and float angle are determined and plotted against steering frequency. The position of the natural frequencies, the banking that occurs and the size of the phase angles can be used as assessment criteria for vehicle agility and stability in the case of dynamic steering excitations. Figure 5 shows such an assessment for an 18-ton truck using the example of lateral acceleration, yaw velocity, and roll angle.

As well as the vehicle parameters which influence the stationary handling, above all the damping properties and the moments of inertia of the vehicle as well as



the dynamic properties of tires, steering system and wheel suspension are crucial to vehicle handling in the case of dynamic driving maneuvers.

Weave test and transition test

The "Weave test" and "Transition test" assessment procedures ([5], [6]) were developed to develop the vehicle's response to small, slow steering movements about the steering zero position. The behavior of the vehicle and steering system during the maneuvers correlates well with the driver's subjective impression of the monitorability of the vehicle in everyday driving.

Transition test

In the transition test the vehicle is turned in, starting from straight-ahead driving in, starting from straight-ahead driving at a constant driving speed, with a slow steering movement (e.g. at 80 km/h with a steering-wheel angle velocity of 5 °/s) to a circular course with a low lateral acceleration of 1 to 2 m/s². In practice this corresponds to the operation of turning at low speed into a curve on a secondary road.

Above all, the time characteristics of the quantities of yaw velocity and steering torque are assessed. Among others, the time after which the vehicle has responded with a quantity of yaw velocity that can be perceived by the driver to the

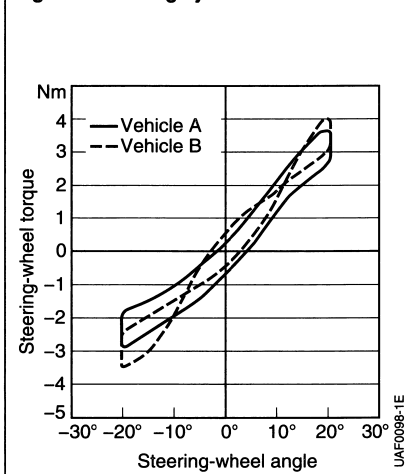
steering excitation is a suitable assessment quantity.

Weave test

In the weave test the vehicle is excited at a constant driving speed (e.g., 80 km/h) with a sinusoidal steering excitation and a slow steering frequency of 0.1 to 0.2 Hz. The maximum lateral acceleration here ranges between 2 m/s² (usual for trucks) and 4 m/s² (usual for passenger cars), i.e., the vehicle is constantly in the linear driving range.

To assess the vehicle handling and steering behavior, the curves for the quantities of steering-wheel torque, yaw velocity, lateral acceleration, and float angle as a function of the input steering-wheel angle are used. The non-linearities (among others friction) in the steering system, tires and wheel suspension give rise to hysteresis loops, like the hysteresis loop of steering-wheel torque vs. steering-wheel angle shown by way of example in Figure 6. The ranges of the hystereses and the rise gradients of the curves are suitable assessment criteria. For example, the steering behavior of a vehicle which demonstrates on overlapping hysteresis loop, like vehicle B in Figure 6, is perceived by the driver to be "indifferent about the center position" and "poor in straight-running stability".

Figure 6: Steering hysteresis in weave test

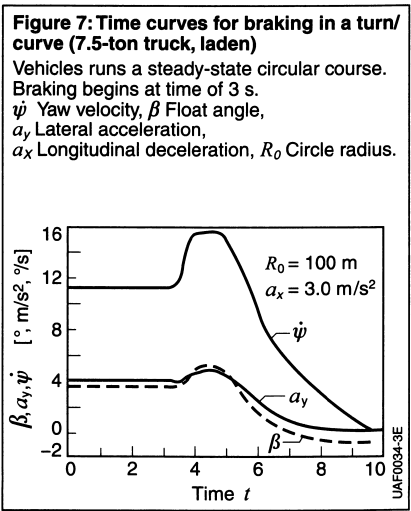


Braking in a turn/curve

The “Braking in a turn/curve” test procedure ([7], [8]) is likewise geared towards a situation that occurs frequently in real driving operation. Starting out from stationary cornering on a circle with a constant radius with a lateral acceleration of between 3 m/s² (usual value for trucks) and 5 m/s² (usual value for passenger cars), the vehicle is braked with a constant steering-wheel angle and a defined deceleration, set by the brake-pedal travel required for this purpose. Several tests are carried out in which the brake-pedal setting and thus the braking deceleration are varied from light braking through to full braking.

Figure 7 shows the time characteristics of yaw velocity, lateral acceleration and float angle of a 7.5-ton truck during a braking operation with a deceleration of 3 m/s². Once the braking has started (time 3.0 s), a clear banking of the driving-dynamics quantities can be made out. It shows that the vehicle turns in to the inside of the curve during the braking operation.

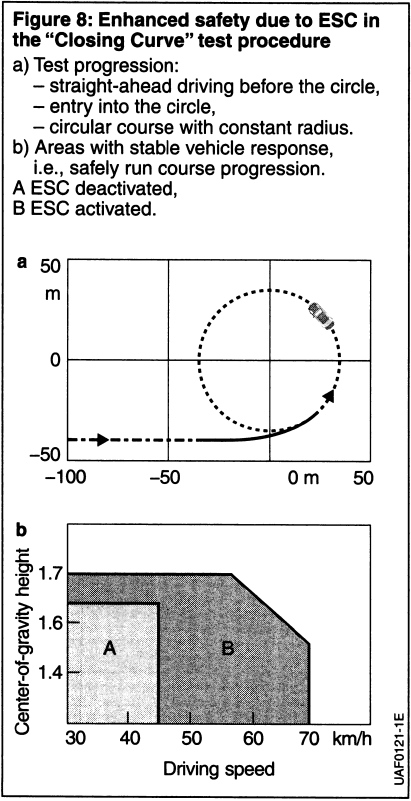
The extent of the banking of the driving-dynamics quantities, which are usually plotted against the varied longitudinal deceleration, are used as an assessment criterion for vehicle stability [18].



As well as the vehicle parameters which influence vehicle handling in the driving maneuvers described up to now, the braking-force distribution between front and rear axles and the design of the brake-control systems ABS and driving-dynamics control (Electronic Stability Control, ESC) have a significant influence on vehicle handling during this maneuver. It is therefore used among other things to tune the brake system and brake-control systems.

Closing curve

The “Closing Curve” test procedure [19] was developed against the backdrop of the change to the ECE regulation ECE-R13, which requires that as of 2014 virtually all heavy commercial vehicles be duly equipped with ESC systems. This test procedure too is based on a real



driving situation, driving into an expressway entrance or exit at too high a driving speed. The enhanced safety afforded by the ESC control system can be gaged with this test. To this end, a trajectory leading rapidly from straight-ahead driving into a curve with a tight curve radius is defined. The course progression is created in such a way that for a certain initial speed a defined increase in the lateral acceleration is obtained, e.g., 2 m/s^2 (Figure 8a).

Several tests with activated and deactivated ESC system and varied entry speeds are conducted. The extent to which the inner cornering wheels lift off is used to assess the enhanced safety afforded by the ESC system in this driving situation (Figure 8b).

References

- [1] ISO 4138 (2012): Passenger Cars – Steady-state circular driving behaviour – Open-loop test methods.
- [2] ISO 14792 (2011): Road vehicles – Heavy commercial vehicles and buses – Steady-state circular tests.
- [3] ISO 7401 (2011): Road vehicles – Lateral transient response test methods – Open loop test methods.
- [4] ISO 14793 (2011): Road vehicles – Heavy commercial vehicles and buses – Lateral transient response test methods.
- [5] ISO 13674: Road vehicles – Test method for the quantification of on-centre handling – Part 1: Weave test (2010), Part 2: Transition test (2006).
- [6] ISO 11012 (2009): Heavy commercial vehicles and buses – Open-loop test methods for the quantification of on-centre handling – Weave test and transition test.
- [7] ISO 7975 (2006): Passenger cars – Braking in a turn – Open-loop test method.
- [8] ISO 14794 (2011): Heavy commercial vehicles and buses – Braking in a turn – Open-loop test methods.
- [9] A. Zomotor: Fahrwerktechnik – Fahrverhalten. 2nd Ed., Vogel-Verlag, 1991.
- [10] M. Mitschke, H. Wallentowitz: Dynamik der Kraftfahrzeuge. 4th Ed., Springer-Verlag 2004.
- [11] ISO 3888: Passenger cars – Test track for a severe lane-change manoeuvre – Part 1: Double lane-change (1999), Part 2: Obstacle avoidance (2011).
- [12] ISO 9815 (2010): Road vehicles – Passenger-car and trailer combinations – Lateral stability test.
- [13] ISO 14791 (2013): Road vehicles – Heavy commercial vehicle combinations and articulated buses – Lateral stability test methods.
- [14] ISO 12021 (2010): Road vehicles – Sensitivity to lateral wind – Open-loop test method using wind generator input.
- [15] ISO 9816 (2006): Passenger cars – Power-off reaction of a vehicle in a turn – Open-loop test method.
- [16] ISO 15037: Road vehicles – Vehicle dynamics test methods – Part 1: General conditions for passenger cars (2006), Part 2: General conditions for heavy commercial vehicles and buses (2012).
- [17] ISO 8855 (2011): Road vehicles – Vehicle dynamics and road-holding ability – Vocabulary.
- [18] E.-C. von Glasner: Einbeziehung von Prüfstandsergebnissen in die Simulation des Fahrverhaltens von Nutzfahrzeugen. Postdoctoral thesis, University of Stuttgart, 1987.
- [19] ISO 11026 (2011): Heavy commercial vehicles and buses – Test method for roll stability – Closing-curve test.

Vehicle aerodynamics

Aerodynamic forces

Vehicle aerodynamics concerns all phenomena that arise as a surrounding medium (air) flows around or through moving vehicles and their components. The drag, forces and moments (see Table 1) are evaluated and usually optimized by changing the shape of the vehicle.

Aerodynamic drag

Aerodynamic drag W_L is given a lot of attention in developing motor vehicles, because it has a direct impact on the vehicle performance and fuel consumption. A critical variable here is the drag coefficient c_W , which describes the aerodynamic quality of the shape of a body in a flow. The aerodynamic drag is also influenced by the frontal area A_{fx} and the dynamic pressure q of the flow, which depends on the air density ρ and the air-flow velocity v .

Dynamic pressure is described by this equation:

$$q = 0.5 \rho v^2.$$

Aerodynamic drag is calculated as (also refer to fluid mechanics):

$$W_L = \frac{\rho}{2} v^2 c_W A_{fx}.$$

Aerodynamic drag is composed of the skin friction drag and form drag:

$$W_L = W_R + W_D.$$

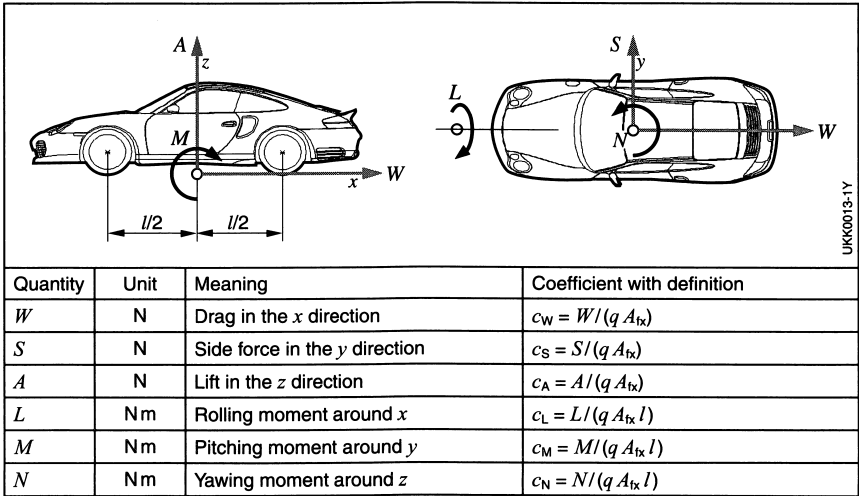
Skin friction drag W_R

Skin friction drag arises due to shear stress τ_w on the walls as the air flows by. Figure 1 shows an example of this on a wing. Skin friction drag is calculated as an integral of the shear stress over the surface:

$$W_R = \int \tau_w \cos \varphi dF.$$

Table 1: Aerodynamic forces and moments

q Dynamic pressure, ρ Air density, v Airflow velocity, A_{fx} Frontal area, c_A Lift coefficient, l Wheelbase.
 $c_{AV} = 0.5 c_A + c_M$ (relative to the front axle),
 $c_{AH} = 0.5 c_A - c_M$ (referred to rear axle).



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Form drag W_D

Form drag arises through pressure changes on blunt bodies caused by shedding. It is calculated as an integral of the pressure over the surface:

$$W_D = \int p \sin \varphi dF.$$

Figure 2 shows the flow pattern for flow around a wing and a hemisphere. Shedding results in vortices, and therefore pressure differences. Form drag is predominant for blunt bodies. For streamlined bodies, the friction component dominates;

Figure 1: Formation of skin friction drag from airflow

v_∞ Airflow velocity,
 φ Angle of impact to the surface element,
 τ_w Shear stress,
 p Pressure,
 dF Surface element.

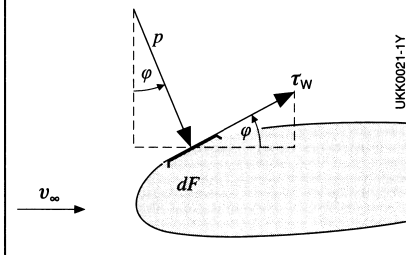
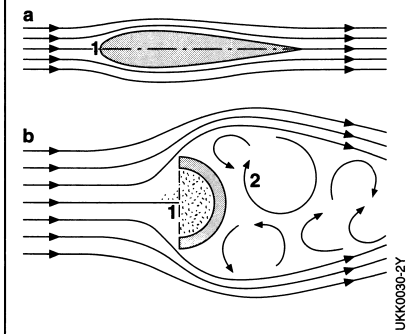


Figure 2: Flow conditions for various bodies

- a) Streamlined body (wing),
 b) Blunt body (hemisphere).
 1 Stagnation point,
 2 Separated flow, vortices.



there is no turbulent shedding from the body's surface.

For the c_W values for different bodies and car body shapes, refer to the various tables (see Drag coefficients, Fluid mechanics and Drag coefficients, Motor-vehicle dynamics).

Induced drag

The difference in pressure above and below a vehicle in flow lead to superimposition of the main horizontal flow around the body. A vertical flow component arises through pressure equalization via the body sides, and wake turbulence develops, which causes the induced drag.

Internal drag

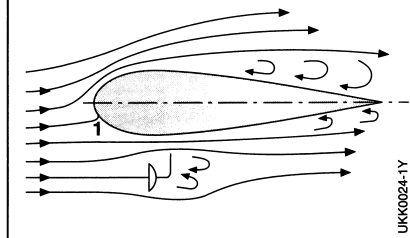
Internal drag arises from pressure loss as air flows through the engine compartment.

Interference drag

Interference drag is produced by the interaction of attachments (such as the wheel suspension, wheels, exterior mirrors, antennas, windshield wipers, spoilers, vanes). The total drag is not the sum of the drag forces of the individual parts, because the mutual interference has to be taken into account. Thus attachments can also cause the c_W value to drop.

The impact of an exterior rearview mirror on the car body is shown here as an example (Fig. 3). It causes the stagnation point to move toward the attachment; this results in asymmetrical flow around the car body. Consequently, there is a greater risk of shedding vortices, and that leads to an increasing c_W value.

Figure 3: Influence of an exterior mirror on a car body (model)
 1 Stagnation point.



Buoyancy force

The curvature of the car's top surface causes the air to flow faster above the vehicle than under it, which generates an undesirable lift force (Fig. 4). This reduces the tire contact forces and thereby has a negative impact on the directional stability.

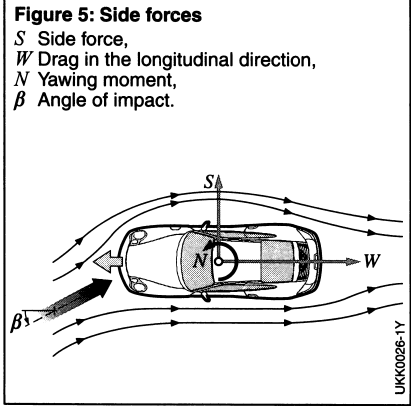
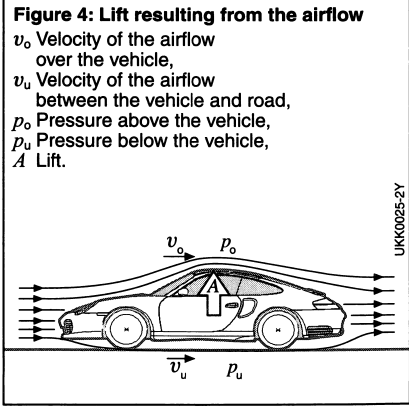
The lift coefficient c_A is the sum of the lift coefficients on the front axle c_{AV} and the rear axle c_{AH} . The difference between the lift on the front and rear axles is referred to as "lift balance" and is a variable influencing directional stability.

The pitching moment M , which acts around the y -axis, is often used as a design variable instead. A positive pitching moment requires understeer, and a negative pitching moment requires oversteer. The rolling moment L around the x -axis is disregarded with respect to aerodynamics.

Lateral force

A car has an almost symmetrical shape when viewed from the front, which means that side forces generated by air flows are small. As soon as the approaching air flow deviates from the x -axis (i.e. in a crosswind), the airflow generates side forces that can affect vehicle behavior considerably (Fig. 5).

The yawing moment N , which acts around the z -axis, is also used as an indicator for crosswind sensitivity. This value is used to derive the yaw-angle velocity and yaw angular acceleration, which provide information about the negative impact of crosswind.



Tasks of vehicle aerodynamics

Aerodynamics has other tasks besides reducing the c_W value (Fig. 6).

Motive power

Impact of velocity on tractive resistance
In order to set the vehicle in motion and keep it moving, the engine has to produce a tractive force Z . This is composed of the aerodynamic drag W_L , rolling resistance W_R , climbing resistance W_S , drag during acceleration W_B and losses in the powertrain W_A . This results in the tractive resistance equation:

$$Z = W_L + W_R + W_S + W_B + W_A.$$

Climbing resistance and acceleration resistance are zero during constant motion on a level stretch. The mileage is substantially determined by the aerodynamic drag, because it increases quadratically with the velocity and surpasses the other tractive resistances as of 80 km/h (Fig. 7).

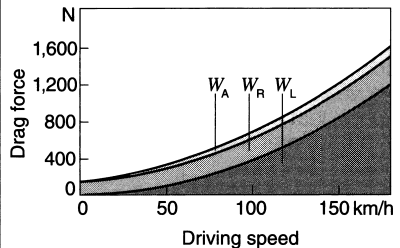
The lower the product of the c_W value and frontal area A_{fx} ($A_{fx} \cdot c_W$ is often called the drag area), the faster a vehicle can move with a suitable transmission ratio. At a constant velocity, a smaller drag area

leads to lower fuel consumption. Consequently, the vehicle produces fewer emissions.

As a function of the low driving speeds in the New European Driving Cycle (average speed of 33.4 km/h, see European test cycle), however, the influence of the aerodynamic drag is greatly underestimated in the manufacturer/consumption data, which can lead to higher fuel consumption in everyday operation for the average driver.

Figure 7: Tractive resistance

W_L Aerodynamic drag,
 W_R Rolling resistance,
 W_A Losses in the powertrain.



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Figure 6: Vehicle aerodynamics: requirements and influences

Vehicle performance

Consumption, emissions, top speed, acceleration, driving dynamics.

Design

Design that is characteristic of the brand, visible or invisible aerodynamic measures (retractable rear spoiler).

Component forces

Doors and cover, windows and sliding sunroof, vibration of mirrors, rear window fluttering, puffing up behavior of convertible top.

Convenience

Draft prevention (mobile roofs), wind noise, prevention of dirt buildup.

Cooling and ventilation

Ventilation of brakes, engine compartment and components, engine cooling, cooling of units (such as gearbox), intercooling, air conditioning, removing condensation from headlights.

Directional stability

Lift forces, lift balance, straight-running stability, lane-change behavior, crosswind stability, handling.

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Directional stability

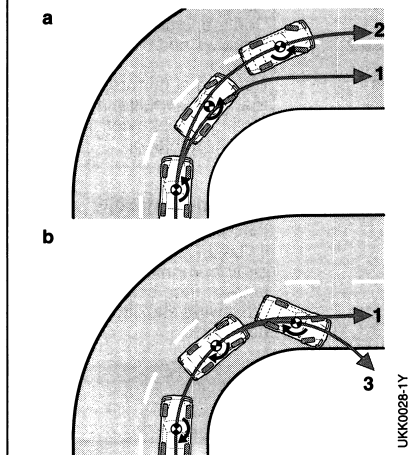
Directional stability, sensitivity of the steering behavior, lane-change behavior, handling and crosswind sensitivity are critically influenced by the lift and side forces. The forces that can be transferred from the wheels to the road in the longitudinal and transverse directions can be no larger than the vertical force (normal force).

The lift acts against the vertical forces resulting from the weight. Consequently, the forces that can be transferred from the tires, above all during motion in a curve, can be reduced until there is a total loss of directional stability, depending on the driving speed and lift. The transferable braking forces also depend on the vertical wheel load, and therefore on the aerodynamic lift.

The task of aerodynamics is therefore to minimize the lift forces through means such as spoilers on the front and rear of a vehicle. For sports cars it can even happen that downforces (i.e. negative lift forces) greater than the weight are produced.

Figure 8: Cornering behavior

- a) Understeering behavior,
- b) Oversteering behavior.
- 1 Correct trajectory,
- 2 Trajectory of an understeered vehicle,
- 3 Trajectory of an oversteered vehicle.



Lift balance

If the lift balance has a lift coefficient c_{AV} on the front axle greater than c_{AH} on the rear axle, there is a positive pitching moment and the design produces understeering driving behavior. Then if the speed in a curve is too high it will be noticed immediately when the steering angle is too high; the vehicle should be stabilized again by backing off the accelerator (Fig. 8a).

If c_{AV} is less than c_{AH} , a negative pitching moment is produced and there is an oversteering design (Fig. 8b). Then if the speed in a curve is too high it will be perceived not through the steering wheel, but through the seat. Backing off the accelerator makes the vehicle unstable; it can be kept in the lane only with targeted countersteering and depressing the accelerator.

Aerodynamic efficiency

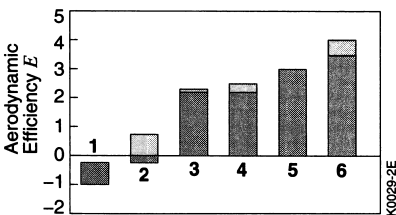
A significant reduction of the lift forces usually comes at the expense of a favorable c_W value. The smaller this impact, the greater the aerodynamic efficiency E . It describes the relationship between the lift and drag coefficients:

$$E = \frac{c_A}{c_W}$$

Figure 9 shows a comparison of the values for the aerodynamic efficiency of different vehicles. Due to the freedom of body design and the very fast racetrack, current Le Mans prototypes have the best

Figure 9: Aerodynamic efficiency for various vehicle types

- 1 Passenger car,
- 2 Sports car,
- 3 Formula 3 racecar,
- 4 Indy car series,
- 5 Formula 1 racecar,
- 6 Le Mans racecar.



efficiency ($E = 3.5$ to 4). Despite a high downforce, a low c_W value is achieved.

Due to the fuel consumption advantages, passenger cars usually have very low c_W values, but positive lift, and therefore a negative efficiency down to $E = -1$.

Cooling and ventilation

The responsibility of aerodynamics consists in requirements-based air intake, so that the heat generated by operating the vehicle is dissipated to the environment (see Engine cooling).

An intensive flow of cooling air can lead to a high internal drag, and therefore increase the aerodynamic drag. Therefore a great deal of attention is given to achieving an optimum balance between cooling requirements and aerodynamics (Fig. 10). To optimize the air intake, air inlets and outlets are located on the vehicle in zones with the greatest possible pressure difference for propulsion. The flow of cooling air can be controlled and adapted by changing the size of the respective openings. The cooling air is conducted through channels to the radiators, which should be designed to be as free of pressure loss and leaks as possible.

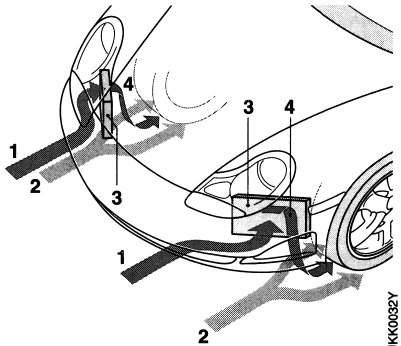
The same criteria also apply to an exhaust air duct up to the air outlet. The air outlet should be positioned downstream with the least possible pressure loss. If it can be placed in a zone where there are already aerodynamic losses, these can be favorably reduced by the arising interference. For example, blowing out the cooling air ahead of the front wheel can

form an air cushion and thereby guide the flow around the wheel with fewer losses, which nearly equalizes the cooling air drag (Fig. 11).

The aerodynamic pressure losses can also be controlled by choosing suitable radiator cores and their dimensions, as well as fans and their frames. To entirely prevent cooling air drag with a low power requirement, electrically closable radiator shutters were introduced in series production back in 1987 (for the Porsche 928). Today this measure is widely used in the premium class.

Figure 11: Interference from blowing out the cooling air ahead of the front wheel

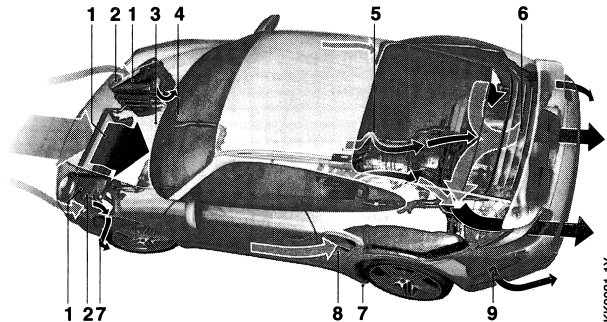
- 1 Airflow approaching the radiator,
- 2 Airflow approaching the wheel,
- 3 Radiator,
- 4 Exhaust flow after the radiator and air cushion ahead of the wheel.



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Figure 10: Cooling tasks for a sports car

- 1 Flow through radiator (engine cooling),
- 2 Air conditioning,
- 3 Front axle gearbox cooling,
- 4 Fresh air intake,
- 5 Rear axle gearbox cooling,
- 6 Engine compartment ventilation,
- 7 Brake ventilation,
- 8 Intercooling,
- 9 Cooling air outlet from the intercooler.



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Comfort and convenience

Aerodynamics helps provide the passengers with comfort primarily by reducing wind noise (aeroacoustics), dirt and drafts.

Aeroacoustics

Wind noise in the passenger compartment of a car is the primary noise source for speeds over 120 km/h; above 200 km/h it is approximately 6 dB(A) louder than all other acoustic sources (Fig. 12). This corresponds to four times the sound power.

The term "aeroacoustics" refers to the production of noise from

- Flow around the body (flow separation, broadband noise from unsteady pressure fluctuations, noise with discrete frequencies such as from antennas and exterior mirrors),
- Component excitation with low-frequency noise excitation due to large surfaces in a flow (such as convertible top, roof, door and cover panels),
- Overflow with noise created by a Helmholtz resonance (such as with open windows, sliding sunroof),
- Leakage due to pressure differences between the passenger compartment and flow around the body as the result of aerodynamic and mechanical component loads.

Improvement of the aeroacoustics

By giving the basic body an aerodynamic shape you can reduce the local flow velocities, influence the flow direction, and prevent or limit turbulent shedding. Generally speaking, whatever reduces the c_w value also leads to lower wind noise.

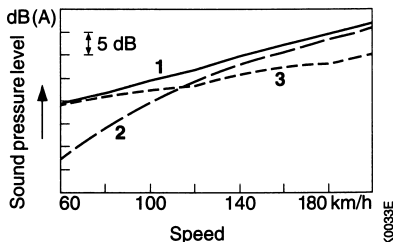
Freestanding attachments (such as mirrors, wipers, antennas, roof transport system) disrupt the flow around the basic body by creating additional flow separation. Therefore they should be positioned where they will produce the least possible disruption and their shape should be optimized. With component excitation, aerodynamics analyses the necessity and placement of reinforcement plates or damping measures.

In the case of flow over cavities (such as an open sliding sunroof), low-frequency pulsing of the air volume in the passenger compartment can occur (humming). That is the resonance case of periodic oscillation of the shear layer between the air overflow and the static air in the passenger compartment (Fig. 13) with the natural frequency of the passenger compartment (Helmholtz resonator principle). As an aerodynamic measure, the shear layer buildup is upset by interference flows, for example, using a wind deflector on the sliding sunroof.

Figure 12: Wind noise components in total noise

Measurement of the sound pressure level with an artificial head in the driver's seat, left ear.

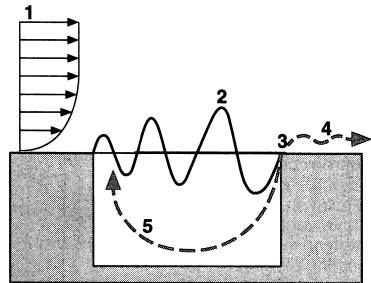
- 1 Total noise,
- 2 Wind noise,
- 3 Other noise.



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Figure 13: Feedback mechanism

- 1 Turbulent oncoming flow,
- 2 Shear layer oscillation,
- 3 Rear edge of cavity,
- 4 Vortex,
- 5 Pressure wave.



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The aeroacoustic quality is evaluated not only by the spectral analysis, but also by taking into account psychoacoustic parameters. These parameters measure higher-frequency noise components; consequently, subjective perception (loudness, sharpness) and speech intelligibility are taken into account better. The sharpness is evaluated independently from loudness and roughness. The articulation index is a measure of speech intelligibility.

Dirt

The wheels of your own vehicle kick up dirt particles, which settle on the surface of your vehicle (self-caused dirtying) or they mix with the turbulent wake and settle on following vehicles (extraneously caused

dirtying). The magnitude and intensity of the contaminated surfaces are caused by local wake turbulence of the surrounding flow. The task of vehicle aerodynamics is to identify such turbulence and reduce it or transpose it to unproblematic zones. Great care is given to avoiding impaired visibility by keeping the windows and exterior mirrors free of dirt buildup.

Draft prevention

Usually there are bothersome drafts when driving with the windows and roof open. This can be clearly observed in a convertible when the top is open, because there is shedding of the flow over a large area behind the windshield frame. A low-pressure region behind the windshield is filled by a backflow. Backflows and turbulence cause the bothersome drafts (Fig. 14).

The vortices can be damped and significantly reduced by the mesh fabric of a windbreak. The task of vehicle aerodynamics is to determine the size and shape of this component as well as its mesh density.

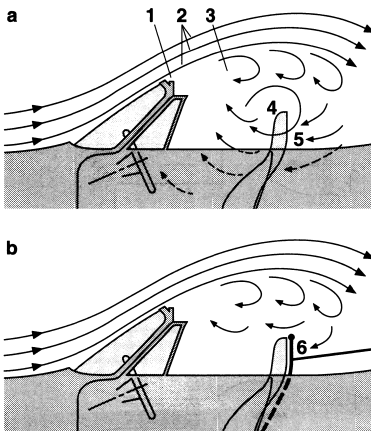
Component forces

Component forces arise from the pressure forces produced by the flow around the body (measured quantity is the c_p value), which vehicle aerodynamics has to determine empirically and increasingly also through calculations. The positive or negative pressure forces acting perpendicular to the outer skin increase along with the airflow velocity and act as a load on the components located there.

In areas where there is shedding (such as A-pillars, exterior mirrors, rear window, rear spoiler), the turbulence results in stochastically varying loads, which have to be taken into account for determining the function and durability of the components.

Figure 14: Reduction of drafts for a convertible by using a windbreak

- a) Vehicle without windbreak,
 - b) Vehicle with windbreak.
- 1 Flow separation at the top edge of the windshield,
 - 2 Flow around vehicle,
 - 3 Unsteady vortex sheet,
 - 4 Vortex direction of rotation in passenger compartment,
 - 5 Airflow between the seats and between the seats and side panels,
 - 6 Windbreak for reducing airflow between the seats.



UKK0035-1Y

Vehicle wind tunnels

Application

Aerodynamic measurements on roads are difficult, because the environmental conditions are neither uniform nor steady. The direction and steadiness of the conditions are constantly changing due to factors such as natural wind, buildings, and traffic. A vehicle wind tunnel is used to represent the air flow affecting a vehicle traveling on the road in an environment that is as realistic and reproducible as possible.

The advantages of using a vehicle wind tunnel as an experimental development tool compared with road measurements are the reproducibility of test conditions, the uncomplicated, reliable, and fast measuring technology, and the ability to disassociate effects (such as driving noise) that occur only in conjunction with road travel. Design prototypes that are not yet road-worthy can be optimized aerodynamically in a wind tunnel with guaranteed secrecy.

Wind tunnel designs

Types of wind tunnel

Aerodynamic parameters are calculated in wind tunnels. They vary in the type of air guidance, test section, size, and road-surface simulation (Table 2). Closed-circuit (closed return) wind tunnels are referred

to as “Göttingen type”, while open-circuit (no return) wind tunnels are referred to as “Eiffel type” (Figure 15). For vehicle aerodynamics, Göttingen wind tunnels with an open test section or with slotted walls are used predominantly.

Standard wind tunnel equipment

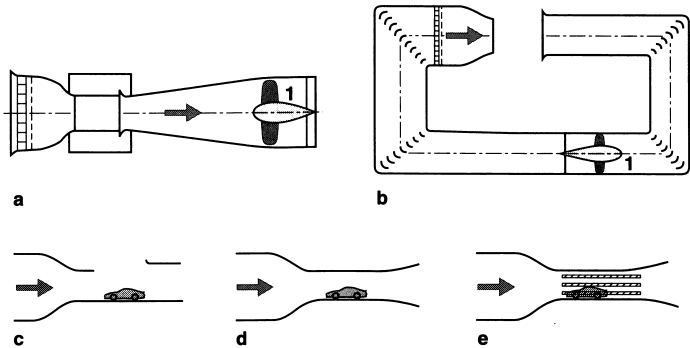
The test section design can be open, have slotted walls, or be closed. It is characterized by the outlet cross-section of the nozzle, the cross-section of the collector, and the length of the test section (Fig. 19, Table 2).

Obstruction $\Phi_N = A_{ix}/A_N$ is also an important parameter. This is the ratio between the frontal area of the vehicle A_{ix} and the nozzle cross-section A_N . On the road, this ratio is $\Phi_N = 0$, therefore it should also be as small as possible in the wind tunnel. Taking the construction and operating costs of a wind tunnel into account, $\Phi_N = 0.1$ is a common figure in practice. This corresponds to a diffuser cross-section of approx. 20 m².

The contraction and contours of a diffuser determine the velocity and steadiness of the wind-tunnel air flow. A large contraction ratio κ of the cross-sectional areas of the prechamber to the nozzle outlet ($\kappa = A_v/A_D$) leads to uniform velocity distribution and low turbulence, as well as to fast acceleration of the airflow.

Figure 15: Wind-tunnel designs

- a) Eiffel type,
 - b) Göttingen type,
 - c) Open test section,
 - d) Closed test section,
 - e) Test section with slotted walls.
- 1 Fan.



The diffuser contour can influence the steadiness of the velocity profile at the diffuser outlet in the test section, and parallelism in relation to the geometrical tunnel axis.

The settling chamber or prechamber is positioned upstream of the nozzle and in the largest cross-sectional area of the wind tunnel. The prechamber contains flow rectifiers, filters, and heat exchangers that improve the quality of the air flow in terms of steadiness and direction, and keep the channel temperature constant.

The fan in most wind tunnels can generate wind speeds of well over 200 km/h. Speeds of this size are only rarely used, for example, for testing the functional safety and stability of body components under wind loads. This is because such tests require the full fan capacity of up to 5,000 kW. Normal measurements are

taken at 140 km/h. At this speed, the aerodynamic coefficients can be determined reliably and at low operating cost. The wind speed is regulated either by changing the fan speed, or by adjusting the fan blades at a constant speed (Table 3).

A wind tunnel balance records the aerodynamic forces at work on the test vehicle and the moments of all four tire contact surfaces; it breaks these forces into x , y , and z components to calculate the aerodynamic parameters.

A wind tunnel balance is normally positioned below a turning platform, which is used to turn the vehicle in relation to the wind direction, and thus simulate the effect of side winds.

In contrast to a real situation, a vehicle in a wind tunnel is stationary and has air blown toward it. The influence of the relative movement between the vehicle and

Table 2: Vehicle wind tunnels in Germany (examples)

Wind tunnel operator	Nozzle cross-section	Collector cross-section	Test section length	Test section design	Simulation of the moving road
Audi	11.0 m ²	37.4 m ²	9.5 to 9.93 m	Open	with 5 moving belts
BMW	18.0 to 25.0 m ²	41.4 m ²	17.9 m	Open	with 5 moving belts
BMW Aerolab	14 m ²	25.9 m ²	15.7 m	Open	1 conveyor belt, width greater than vehicle width
Daimler	28.0 m ²	61 m ²	19 m	Open	with 5 moving belts
IVK Stuttgart	22.5 m ²	26.5 m ²	9.9 m	Open	with 5 moving belts
Ford	20.0 m ²	28.2 m ²	9.7 m	Open	—
Porsche	22.3 m ²	37.7 m ²	13.5 m	Slotted walls, open	—
Volkswagen	37.5 m ²	44.8 m ²	10.0 m	Open	—

Table 3: Wind-tunnel fans

Wind tunnel	Drive power	Fan diameter	Wind velocity	Control
Audi	2,600 kW	5.0 m	300 km/h	Revolutions per minute
BMW	4,140 kW	8.0 m	300 km/h	Revolutions per minute
BMW Aerolab	3,440 kW	6.3 m	300 km/h	Revolutions per minute
Daimler	5,200 kW	9.0 m	265 km/h	Revolutions per minute
IVK Stuttgart	3,300 kW	7.1 m	260 km/h	Revolutions per minute
Ford	1,950 kW	6.3 m	185 km/h	Revolutions per minute
Porsche	2,600 kW	7.4 m	220 km/h	Revolutions per minute
Volkswagen	2,600 kW	9.0 m	180 km/h	Blade adjustment

the road cannot be taken into account. Therefore, in addition to wind tunnels with a stationary floor, there is now increased use of wind tunnels with moving belts incorporated into the floor to imitate motion of the car over the road and rotating wheels (Figure 16). The quality of the air flow between the vehicle and the road can thus be improved and made significantly more realistic.

Wind-tunnel auxiliary systems

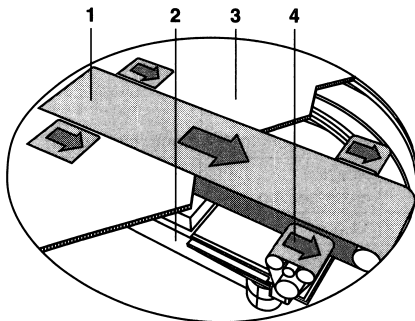
Frontal-area measuring system

The frontal-area measuring system (laser or CCD system) measures the frontal area of the vehicle by optical means. The results are then used to calculate the aerodynamic coefficients from the forces measured in the wind tunnel.

The ideal pressure transducer system of a wind tunnel can simultaneously record the pressure curve at up to 100 pressure measuring points, for example, using the flat sensors affixed to the car body surface for the pressure distribution. The miniature pressure sensors used for each measuring point have no wearing parts

Figure 16: Turning platform on test section floor with flat belts incorporated

- 1 Moving belt between the wheels,
- 2 Balance,
- 3 Turning platform,
- 4 Wheel drive unit with small moving belt.



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Figure 17: Measurement of pressure distribution (application example)

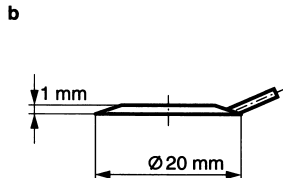
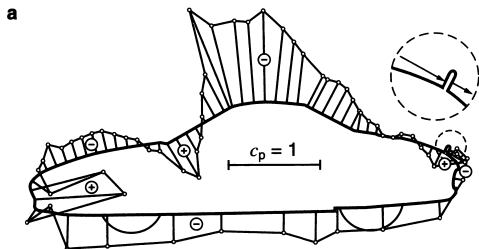
- a) Pressure distribution on midsection of vehicle surface ($y = 0$), measured using 63 flat sensors,
- b) Flat sensor.

The lines show the static pressure acting perpendicular to the measurement point. The arrow length indicates the magnitude of the pressure.

The line under $c_p = 1$ shows the scale.

c_p is a dimensionless pressure coefficient; it describes the pressure p_{∞} at any point on the vehicle at the velocity v_{∞} with respect to the dynamic pressure p prevalent at this velocity.

$$c_p = \frac{p - p_{\infty}}{0.5 \cdot \rho \cdot v_{\infty}^2}$$



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and can therefore electronically scan at a high frequency (Figure 17).

Traversing cradle

The traversing cradle allows the measurement of the complete vehicle airflow field. Each position in the test section can be located by coordinates and reproduced. According to the sensors present at each point, the pressure, velocity, or noise sources at each location can then be determined.

Smoke tubes

Smoke tubes are used to visualize the airflow, which would otherwise be invisible. The smoke pattern enables detection of flow separation, the vortices of which worsen the coefficients (Fig. 18). The non-toxic “smoke” is usually created by heating a glycol mixture in an oil-vapor generator. Other methods for visualizing the air flow include:

- Tufts on the vehicle surface,
- Tuft sensors,
- Photographs of flow patterns with quick-drying liquid mixture of kerosene or talc,
- Helium bubble generator,
- Laser sheet.

Pollutant dispersion system

The pollutant dispersion system can be used to spray the vehicle in the wind tunnel with water ranging from light mist to heavy rain. The flow patterns can be visualized and documented by mixing chalk or a fluorescent agent to the pollutant.

Hot-water unit

The hot-water unit provides heated water at a constant rate for measuring radiator cooling capacity in prototypes that are not yet roadworthy.

Wind tunnel variants

A vehicle wind tunnel is a major investment. This investment, combined with high operating costs, make it an expensive test environment with a high hourly rate. Only very frequent utilization of vehicle wind tunnels for aerodynamic, aeroacoustic, and thermal experiments can justify the construction of several wind tunnels specialized for different tasks.

Model wind tunnel

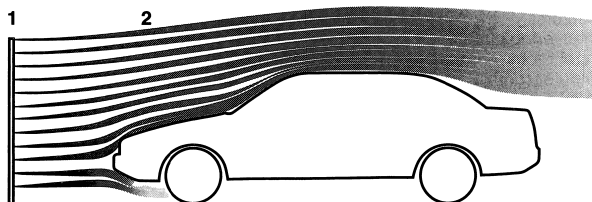
A model wind tunnel can considerably reduce operating costs, due to the lower construction requirements and technical complexity. Depending on the scale (1:5 to 1:2), it is easier, faster, and more cost-effective to change the shape of vehicle models.

Model tests are primarily used in the early development phase for optimizing the basic aerodynamic shape. With the support of designers, plasticine models are used in a wind tunnel to optimize shapes or develop complete alternative shape variants and establish their aerodynamic potential.

Once there is a data set, new production methods (such as rapid prototyping) can be used to quickly produce models to any scale with accurate details. This allows for important investigations of the models in order to optimize details, even after the shaping phase.

Figure 18: Use of a smoke rake consisting of smoke tubes to make the airflow visible

- 1 Rake,
2 Smoke stream.



Acoustic wind tunnel

In an acoustic wind tunnel, extensive sound insulation measures mean that the sound pressure level is approximately 30 dB (A) lower than in a standard wind tunnel. This provides a sufficiently large signal-to-noise ratio of more than 10 dB (A) from the useful signal of the vehicle to allow identification and evaluation of wind noises generated by the air circulation and throughflow.

Climate-controlled wind tunnels

Climate-controlled wind tunnels are used for thermal analysis and protection of vehicles in defined temperature ranges at different load conditions.

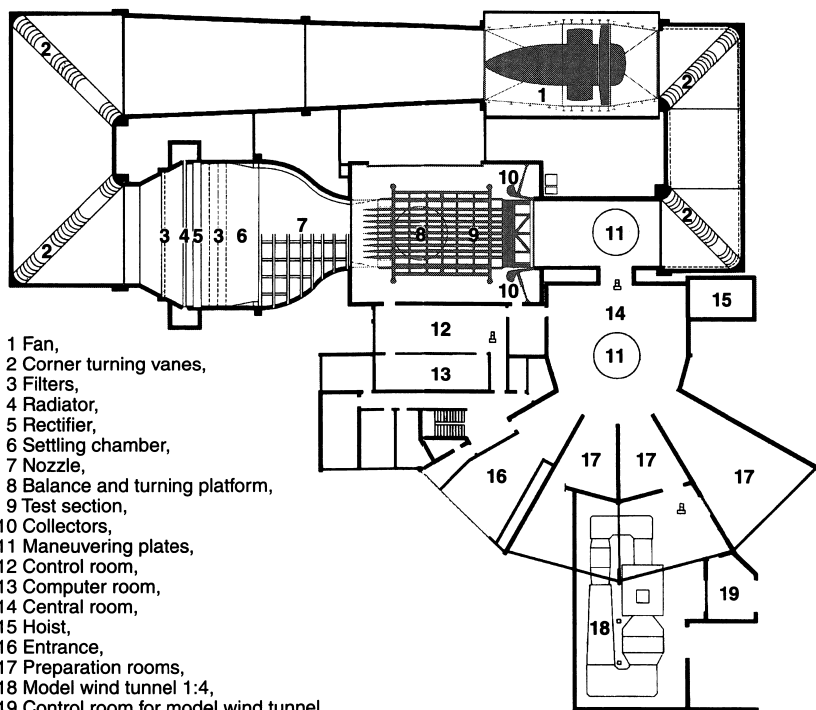
Large-scale heat exchangers are used to set a temperature range, e.g. from -40°C to $+70^{\circ}\text{C}$ at a low control tolerance limit of approx. $\pm 1\text{ K}$.

The vehicle is positioned on dynamometer rollers and driven at the required load condition or load cycle. Wind speed and roller speed must be exactly synchronized, even at low speeds. If necessary, uphill or downhill gradients can be simulated to introduce realistic road factors to the vehicle operating cycle.

There is also the option of regulating the air humidity or simulating solar radiation (using a bank of lamps).

The extensive aerodynamic challenges in vehicle design are not all covered by enhanced test-bench simulations as described above. In addition to experimental testing, manufacturers are therefore making increasing use of CFD models (Computational Fluid Dynamics). They allow preliminary decisions to be made to reduce the test workload.

Figure 19: Vehicle wind tunnel (example of the Göttingen type from Dr. Ing. h.c. F. Porsche AG)



Vehicle acoustics

Vehicle acoustics deals with all vibrational excitations and their transmission in the vehicle and in airborne noise. The primary sources here are the engine, excitation of the tires and chassis due to the texture and irregularities of the road, as well as excitation due to the airflow around the vehicle with increasing driving speed.

Table 1: Limits in dB (A) for noise emission from motor vehicles as per [1]

Vehicle category		since October of 1995 dB (A)
Passenger cars		
With spark-ignition or diesel engine		74
– with direct-injection diesel engine		75
Trucks and buses		
Permissible total weight below 2 t		76
– with direct-injection diesel engine		77
Buses		
Permissible total weight 2 t to 3.5 t		76
– with direct-injection diesel engine		77
Permissible total weight above 3.5 t:		
– engine power output up to 150 kW		78
– engine power output above 150 kW		80
Trucks		
Permissible total weight 2 t to 3.5 t		76
– with direct-injection diesel engine		77
Permissible total weight above 3.5 t (German Road Traffic Licensing Regulations: above 2.8 t):		
– engine power output up to 75 kW		77
– engine power output up to 150 kW		78
– engine power output above 150 kW		80
Off-road and four-wheel drive vehicles		
These vehicles are governed by higher additional limit values for engine brake and compressed-air noises.		

Legal requirements

Noise emission from vehicles and its testing

Test procedures used to monitor compliance with legal requirements are concerned exclusively with exterior-noise levels. The EC Directive 70/157/EEC enacted in 1970, together with its last revision 2007/34/EC [1], defines measurement procedures for noise emissions from stationary and moving vehicles, as well as limit values for noise emissions from moving vehicles (Table 1) for various vehicle categories. On account of the inadequate effectiveness of these statutory provisions in real traffic, the measurement method has been revised with the aim of better reproducing urban traffic in the test procedure.

The new EU noise legislation – Regulation 540/2014/EC [2] – has been in force since April 2014 and its application has been mandatory for new homologations of vehicles since July 2016. The UN Regulation UN R51.03 [3] applied in parallel in the EU has been in force since January 2016 and follows the same timeline as the EU Regulation. For the old legislation there are transitional provisions until July 2022 (until July 2023 for N2 category vehicles).

In addition, tread noise requirements were defined in the regulation 661/2009/EC [4]. Depending on the tires, the test area is 70 or 80 km/h.

Measuring noise emissions from moving passenger cars and trucks up to a permissible total weight of 3.5 t
The vehicle approaches line AA, which is located up to 10 m from the microphone plane, at a constant velocity of 50 km/h (Figure 1). After the vehicle reaches line AA, it continues under full acceleration as far as line BB (placed 10 m behind the microphone plane), which serves as the end of the test section. Passenger cars with manual transmissions and a maximum of four forward gears are tested in second gear. Consecutive readings in second and third gear are employed for

vehicles with more than four forward gears, while sports cars are tested in third gear in accordance with the definition in the Directive. Vehicles with automatic transmissions are tested in the D position.

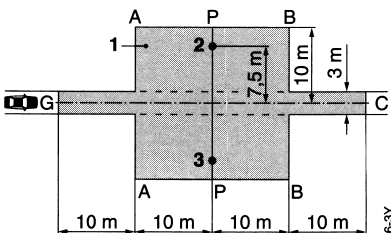
The noise-emissions level is defined as the maximum occurring sound level which is measured on the left and right sides of the vehicle at a distance of 7.5 m from the center of the lane. In a test in two gears, the noise-emissions level is the arithmetic mean from the measurement of both gears.

Measuring noise emissions from moving trucks starting from a permissible total weight of 3.5 t

The vehicle approaches line AA, which is located up to 10 m from the microphone plane, at a constant velocity (Figure 1). After the truck reaches line AA, it continues at full acceleration as far as line BB (also placed 10 m from the microphone plane), which serves as the end of the test section. The driving speed adopted is dependent on the gear tested and the rated engine speed. The choice of gears selected is based on the fact that at least the rated speed must be reached within the test section, however, the vehicle must pass through the test section without reaching the engine-speed limitation. The noise-emissions level is defined as the maximum occurring sound level from all the measurements.

Figure 1: Test layout for driving-noise measurement

- 1 Road surface as specified by ISO 10844 [5],
- 2 Left microphone,
- 3 Right microphone.



Measuring noise emissions from stationary vehicles

Following noise approval of a vehicle, measurement of noise emissions from stationary vehicles makes it possible to determine a reference noise value for the noise emissions of the vehicle in traffic when in "proper" condition. This makes it easier for the vehicle's noise emissions to be checked, for example, by the police or regulatory official. Noise emission from stationary vehicles is measured at a distance of 50 cm from the exhaust outlet at a horizontal angle of $45^\circ \pm 10^\circ$ relative to the direction of exhaust flow. During the measurement, the engine has to be brought up to a speed dependent on its nominal speed. Upon reaching this test speed, it is to be maintained for 3 sec. and then the throttle response switch is to be quickly moved to the idle position. The maximum A-weighted sound pressure level determined during this measurement is entered in the vehicle documentation with the suffix P' (to distinguish it from data of earlier test procedures).

A maximum deviation of 5 dB from the entered reference value is permitted for tests in traffic.

Minimum noise from vehicles

When quiet electric motors began being widely used as a drive source for vehicles, it became a problem for pedestrians: They have great difficulty recognizing these vehicles in traffic in a timely manner. Investigations have shown that this is particularly the case in the low-speed range up to about 20 to 30 km/h. To reduce the danger of collisions between vehicles and pedestrians, work is currently being done to specify an international standard for minimum noise emissions. These provisions are mandatory from July 2019 for the new homologation of vehicles and mandatory from July 2021 for all newly approved vehicles.

Development work on vehicle acoustics

Measuring equipment for acoustics

The measuring techniques listed in the following are used to measure and evaluate not only the exterior noise level, but also the interior noise level and in particular the various vehicle vibrations [6]:

- Sound-pressure recording is performed with capacitor microphones, e.g. using sound-level meters, in dB(A).
- The dummy-head measurement technique enables noise to be recorded by microphones built into the ears of an artificial head. Artificial- or dummy-head technology provides the opportunity of making realistic recordings, so that they can be compared with other similar recordings at a different place and time.
- Measuring rooms for standard sound measurements are generally equipped with highly sound-absorbent walls. Depending on the objective, these rooms can be equipped with chassis dynamometers, so that a vehicle can be evaluated even while in driving mode.
- For vibrations and structure-borne noise acceleration, sensors (mass partly under 1 g) are used which operate, for example, according to the piezoelectric principle. Laser vibrometers are used for rapid, non-contact measurements based on the Doppler principle.
- Position measuring devices are used, for example, to record movement of the shock absorbers and chassis relative to the body.
- Vibration test benches provide the option of measuring the various cases of vibrational excitation in the overall vehicle, the body, or individual components. Hydropulsers permit excitation of low frequencies smaller than 150 Hz with large vibration paths for simulating road excitation. Shakers, on the other hand, permit high frequencies up to 2 kHz for simulating engine excitation.

- Ride comfort segments simulate various kinds of road excitation, ranging from different kinds of asphalt to man-hole covers or concrete transverse joints.
- Reverberation rooms enable the sound insulation of individual materials or components to be determined, for example, for doors or whole bulkheads in the engine compartment.

Calculating methods in acoustics

Vibrations and oscillations

Natural-vibration calculations are performed using the finite-element method (FEM). Model adjustment with experimental modal analysis or modeling of forces acting during operation enables the calculation of real operational vibration shapes. In this way, structures can be optimized in early design phases with regard to their vibration response and sound radiation.

Airborne noise and fluid-borne noise

Sound-field calculations, for example of cabinet radiation or in cavities, are carried out using the finite-element method (FEM) or boundary-element method (BEM) ([7]).

Ride comfort simulation

Effects of vibration from the road on the chassis and body can be simulated by using rigid body models in combination with the road excitation. What is interesting here is that it is also possible to connect the drive unit via the engine mount and transmission mount. Such models enable variable coordination of the chassis mount and power train mount, typically in the frequency range up to 50 Hz.

Sources of noise

Engine and transmission acoustics

Engine noises are caused primarily by the mechanical movement of components. Even the combustion process in the cylinders causes excitations of the engine structure. In all cases, the vibrations are, on the one hand, directed through the structure and, on the other hand, radiated as airborne noise. In the process, noises can also occur dominantly at a clear distance from the excitation location. This is determined by the sensitivity of the membrane surfaces and the natural-vibration response of the local structure.

Important sources of noise on the engine are the crankshaft drive, the camshaft drive, the crankcase, the oil pan, the cylinder-head cover, the auxiliary power take-offs (e.g. alternator and water pump), and the belt drives. All of these elements together shape the basic acoustic character of the unit (engine/transmission combination).

The objective of the acoustician is to achieve harmonization of the vibration and noise behavior in the power train by means of suitable technical modifications. Some of the important influencing factors to take into account include the following:

- Tolerances, clearances and structures,
- Thermal expansion in clearances,
- Avoiding resonances,
- Choice of materials,
- Vibration shapes of the engine/transmission unit,
- Muffler design.

Here, most of these are in a conflict of interests with other development disciplines, particularly space design (package), energy management and driving dynamics. Thus, in a four-cylinder engine, the often dominant second engine order, which is frequently perceived as humming, can be significantly reduced with balancer shafts and very rigid units. But this comes at the expense of additional

weight, more complicated structures and a considerable dip in performance.

Engine and transmission acoustics also has the task of largely reducing the transmission of rigid-body sound vibrations to the body and the airborne noise. Starting points for this are the design of the power train mount with respect to insulation, low-vibration designs, short, compact connections of auxiliary units, and prevention of large, smooth vibration surfaces by means of ribbing or additional damping materials.

The engine mount deserves special attention. Stiff assembly bearings improve driving dynamics, but convey to the driver a very raw engine noise and increase the vibration input into the body. If the bearings are too soft, it means the engine can move large distances, which can lead to unpleasant bucking motion of the whole vehicle. Highly insulating, hydraulically damped elements can be used to bring about a certain softness with good vibration comfort for the whole vehicle.

Gas exchange noise (exhaust and intake system)

The energy transmitted by the combustion processes to the inducted air is output in time with the ignition processes and the valve movements via the exhaust-gas system to the surroundings again. The design of the exhaust manifolds, the exhaust pipes and the mufflers decisively affects which frequency components are radiated with which levels. Here, the sound output, not only at the orifice but also at the surface of all the components, is important. The noise dynamics of this acoustic source are enormous, easily reaching 15 dB and higher when the level difference is considered between idle and full load. On the basis of these dynamics, this source is ideally suitable for sound designs.

Chassis acoustics

Excitation from the road introduces vibrations into the chassis. These are strongly attenuated at first by the damping of the tires, but vibrations can still be transmitted in the chassis and into the body. Moreover, the road can incite natural resonance in the chassis, which can reach the passenger compartment in the form of unpleasant droning if the body has a certain sensitivity. Chassis acoustics has the task of finding an optimum acoustical configuration of the chassis without impairing the dynamic handling characteristics of the vehicle.

Body acoustics

The body is the interface between the chassis, power train and mounts of many auxiliary drives. These include, for example, fans, servomotors and even loudspeakers. The body, however, is also a point of attack for airflow forces. It surrounds the driver and passengers and protects them from external influences. But it also transmits vibrations and introduces them to the passenger compartment, where the occupants can both feel and hear them.

Body acoustics deals specifically with minimizing the transmission of sound and preventing airborne noise excitations, which, along with resonance in the passenger compartment, can lead to unpleasant droning noise. It also deals with optimizing the natural-oscillation characteristics of the body, because the body's eigenmodes – such as the torsion of the basic body or the first bending mode of the front end – can lead to shaking of the vehicle, which feels like when the vehicle is very fluttery on the road.

High rigidity of the attachment points for the chassis and engine provide for low excitation, sealing the byways reduces the conduction of sound within the body, and damping materials provide for sufficient insulation.

Body acoustics also includes aeroacoustics, in which optimized transitions at vehicle edges and flows around components should prevent turbulent flows from developing. These can make themselves apparent as unpleasant interior and exterior noise at high speeds.

Ride comfort

Ride comfort concerns all of the ride-related components that the driver and passengers touch. In particular these include the steering wheel, shift lever, pedals, and of course the seat. Excitations can come from the road, body, or engine and are in the frequency range under 50 Hz. The objective is to minimize all vibrations.

Dynamic strength

All vehicle components are subject to vibrational excitation, and therefore to dynamic loads. To ensure that all components have a sufficient dynamic load capacity over a long time period, they have to be investigated with respect to their dynamic strength. To do so, individual components as well as entire assemblies are subjected to continuous loading on test benches by electrodynamic shakers or hydraulic punches. The excitation profiles are gained from load measurements from real driving profiles. If necessary, the components can also be tested with climatic adaptations, such as low or extremely high temperatures. Exhaust systems can, for example, have hot gases pumped through them to simulate the engine exhaust gases. These tests provide valuable indications for optimizing details, as a result of which materials, flanges or ribs are often optimized again.

Sound design

Definition of sound design

Sound design is the active and specific matching of a product to a desired noise behavior. The aim is to lend the product a typical "sound" expected by the customer and corresponding to the brand identity and to arouse quite specific emotional associations with the product.

The most well-known field is the automobile sector, where both customers and the press show great interest in the result of the sound development of products. However, intensive work is also being carried out on the sound of products in many other fields, such as, for example, household appliances (vacuum cleaners and washing machines) and even foods (the "crunch" of potato chips).

Terms other than sound design are also commonly encountered. Sound engineering is, as a rule, the concrete technical realization on the product, i.e. the specific technical implementation of a target sound elaborated in sound design. Sound cleaning is the suppression of unwanted noise to achieve an optimally neutral noise behavior. This is often carried out for luxury vehicles as a starting point for developing a discreet sound.

Sound at the vehicle

The steps for implementing sound at a product, are generally speaking, always the same. For clarification purposes, this is explained by reference to the automobile sector.

Sound is the audible feedback of the vehicle at a specific operating state arrived at by the driver using the accelerator pedal and choosing a gear. Sound is accordingly described by the engine's operating point with regard to engine revs and load, as well as their time variance – i.e. acceleration, steady-state driving or deceleration. Starting and idle noises are added.

Depending on the design, the subjective perception of the throttle response is "spontaneous" or "sluggish", regardless of the real physically measurable acceleration.

A distinction is made between a vehicle's interior and exterior noises. Exterior noise is shaped by the airborne noise radiation of all the components, primarily by the engine and the exhaust-gas and air-intake systems and their orifices. In addition to these, there is disruptive noise from rolling and the wind. The acoustic conditions in the passenger compartment are additionally influenced by the structure-borne noise and the acoustic transfer paths through the body.

Sound quality

Sound quality refers to discrete components that can be actively moved or changed, such as switches, levers, servomotors, or flaps. Here, the actuating noise is intended to underline the quality and the soundness of the product. Opening and closing doors and hoods also has to sound good and must not be jarring. Moreover, sound quality also includes prevention of rattling, grinding and squeaking noises produced by components touching and rubbing against each other.

Assessment methods

Psychoacoustics

Sound cannot be assessed with the standard assessment method based on loudness, roughness or tonality. When the sound of a vehicle is mentioned, for the most part terms such as "sporty", "sonorous", "aggressive" or "inspiring" are used. These must be translated into a technical language so that the development engineers know at a later stage what spectral content, what dynamics and what time variances are responsible for them. For this reason, these subjective factors are being increasingly brought into play and

structured in order to establish assessment criteria from them. In the practical course of a psychoacoustic assessment (Figure 2), sound samples are assessed according to antagonistic value pairs (such as good – bad, loud – quiet, weak – strong) so as to arrive at a characterization of the sound.

Virtual prototyping

Analyses and measurements conducted on components can be used to make predictions about sound behavior and the leeway for making modifications. Vehicle measurements are also made using artificial-head technology. Variations can then be simulated in a narrow range with modern computer technology, always with a view toward whether a “virtually created sound” can also be technically realized.

Now the virtual sounds processed using computer technology can be evaluated by test subjects with respect to previously developed evaluation criteria. From this, it is possible to derive assessor preferences, which are now, however, techni-

cally stored and provide indications as to which frequency elements are necessary or offending and which components come into consideration.

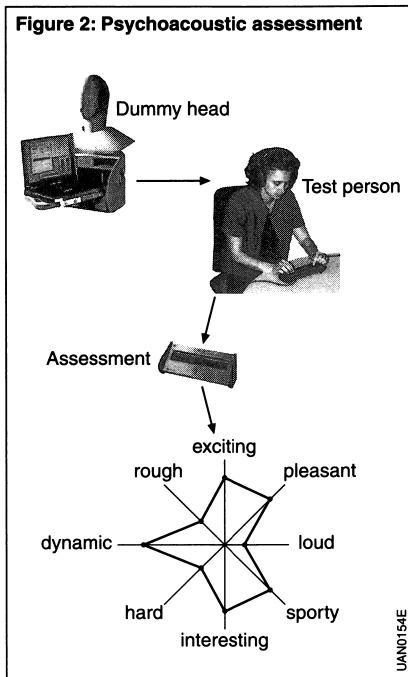
Active sound

Of course, missing sound elements can also be provided with active speaker systems inside and outside the vehicle. This technology is currently gaining ever more importance, because electric vehicles and hybrid vehicles in electric driving mode are frequently evaluated as being too quiet. Significant improvements can be achieved here by adding electronic sound. These provisions are mandatory from July 2019 for the new homologation of vehicles and mandatory from July 2021 for all newly approved vehicles (see Minimum noise from vehicles).

References

- [1] Commission Directive 2007/34/EC of 14 June 2007 amending, for the purposes of its adaptation to technical progress, Council Directive 70/157/EEC concerning the permissible sound level and the exhaust system of motor vehicles.
- [2] Regulation (EU) No. 540/2014 of the European Parliament and of the Council of 16 April 2014 on the sound level of motor vehicles and of replacement silencing systems, and amending Directive 2007/46/EC and repealing Directive 70/157/EEC.
- [3] UN R51-03: Uniform Provisions Concerning the Approval of Motor Vehicles Having a Least Four Wheels with Regard to their Sound Emissions.
- [4] Regulation (EC) No 661/2009 of the European Parliament and of the Council of 13 July 2009 concerning type-approval requirements for the general safety of motor vehicles, their trailers and systems, components and separate technical units intended therefor.
- [5] ISO 10844:2011: Acoustics – Specification of test tracks for measuring noise emitted by road vehicles and their tyres.
- [6] Klaus Genuit: Sound-Engineering im Automobilbereich. 1st ed., Springer, 2010.
- [7] L.C. Wrobel, M.H. Aliabadi: The Boundary Element Method. 1st ed., Wiley, 2002.

Figure 2: Psychoacoustic assessment



Chassis systems

Definition and design

Aside from the drive (engine, transmission) and the vehicle body (body, interior), the chassis constitutes one of the classic, property-determining main assemblies of a motor vehicle. As the link between the vehicle body and the road, the chassis is crucial to generating and transmitting the horizontal and vertical forces between the tires and the road which make it possible to propel, brake, and steer a vehicle. Furthermore, it is the decisive factor in determining the vibration characteristics that occur while driving and vibration effects in the longitudinal, lateral, and vertical vehicle direction. The chassis and its properties therefore have a significant impact on a vehicle's driving dynamics, driving comfort, and driving safety.

Because of the many and sometimes contrary requirements imposed on a vehicle between the conflicting priorities of driving dynamics, driving comfort, and driving safety, the design and tuning of a chassis are subject to high levels of complexity.

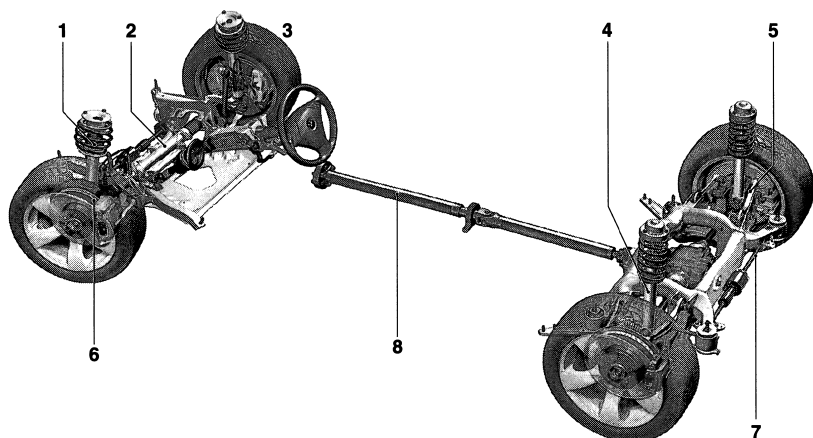
A chassis in the classic sense is divided into the following subsystems: Suspension

(body suspension and anti-roll bars), vibration dampers, wheel suspension and wheel control, wheels with tires, steering, and brake system. Figure 1 shows a typical chassis and its components. The fundamentals and particulars of the individual subsystems of a chassis will be addressed in greater detail in the following.

Where the chassis in the past was characterized above all by mechanical components, so today active components and networked chassis systems are – above all in the passenger-car sector – state of the art. The interaction of active components, sensors for monitoring and calculating the driving condition, and intelligent control approaches open up a multitude of new possibilities for influencing relevant chassis properties in response to specific situations. These possibilities for the individual subsystems will also be addressed in greater detail in the following.

Figure 1: Components of a passenger-car chassis (picture: BMW)

- 1 Body suspension, 2 Steering, 3 Wheel with tire, 4 Body vibration damper,
- 5 Wheel suspension, 6 Brake system, 7 Anti-roll bar, 8 Propeller shaft (part of drivetrain).



Basic principles

The roads normally used by motor vehicles feature irregularities that cover a frequency range of up to around 30 Hz and are the most intensive source of excitation in the vehicle. The resulting excitation leads to vertical movements (vertical accelerations) of the vehicle and its occupants. The interface between the road surface and the vehicle is the chassis system, the main task of which is power transmission between the environment and the vehicle body. This means that both the dynamic characteristics of the vehicle and driving safety, as well as driving comfort, are heavily influenced by the choice of chassis system.

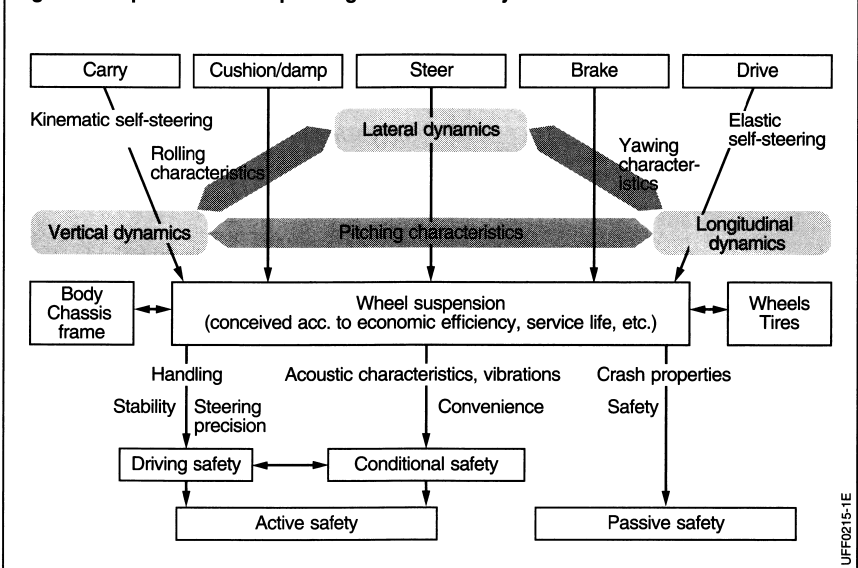
The dynamics of the vehicle response, however, are not exclusively determined by the chassis system components, but rather are much more a consequence of the combination of different overall vehicle parameters. As a rule, influencing the vehicle dynamics with measures on the chassis system involves high complexity, in particular because the effects of parameter variations have to prevail among the conflicting priorities of driving safety and driving comfort.

Driving safety depends decisively on the contact relationships between the tires and the road surface, and thus on the longitudinal and lateral forces that can be transferred. A fundamental aim in chassis system configuration with regard to driving safety is therefore always to minimize the dynamic wheel load fluctuations that cause a reduction in the level of force that can be transferred.

The driving comfort, on the other hand, depends on the movements and accelerations that affect the occupants (above all in the vertical direction). Depending on the area of application, comfort is of great relevance and should in no way be regarded merely as a concomitant of system development. In the case of professional drivers in particular, adequately high driving comfort is to be ensured in order to prevent long-term damage to health. The effective value of the body acceleration has proven to be a good evaluation parameter in this context.

These core parameters, however, initially only describe the potentials of a chassis system to meet the corresponding requirements. The actual dynamics of the vehicle, the driving safety, and the comfort also

Figure 2: Requirements for a passenger car chassis system



depend decisively on the choice of road surface parameters (environment) and vehicle-internal manipulated variables (for example, steering angle and accelerator pedal position) set by the driver. The fundamental requirements for a chassis system are shown in Figure 2 [1]. The chassis as a system (without electronics) is traditionally divided into the following subsystems:

- Suspension,
- Shock absorbers,
- Wheel suspension and suspension linkage,
- Tires,
- Steering,
- Brake system.

Road excitation

Before the basic principles of individual components of the “chassis” as a system are presented below, first of all the quantification of the road excitation will be discussed and then the effect of the excitation on the vibrating overall “vehicle” system. Alongside the adaptation of the oscillatory

system to the external road excitations, vehicles also require the influence of internal sources of excitation on the vibration characteristics (internal-combustion engine, wheel, and tires) to be minimized [2].

Knowledge and a description in the form of objective variables of the road excitation that causes vibrations are required in order to examine the vibration characteristics and to configure the suspension/shock absorber system of a chassis. Whereas minor irregularities can already be compensated for by the suspension characteristics of the tire, an element between the wheels and body that changes its length is required to reduce greater body movements. Steel springs are used most frequently here, delivering a return force that depends on the change in length. The result is – taking account of the wheel and body masses – a system that can vibrate and requires other elements for damping.

Courses of irregularity

When describing courses of irregularity in a road depending on the path or time, a distinction is made between harmonious (sinusoidal), periodical, and stochastic courses of irregularity (Figure 3).

Harmonious course of irregularity

In this case (Figure 3a), the irregularity height h following distance driven x or point in time t results in:

$$h(x) = \hat{h} \sin(\Omega x + \varepsilon) \quad (\text{Equation 1}),$$

$$h(t) = \hat{h} \sin(\omega t + \varepsilon) \quad (\text{Equation 2}),$$

where:

$\hat{h}$ Amplitude of irregularity height,

L Period length,

v Speed,

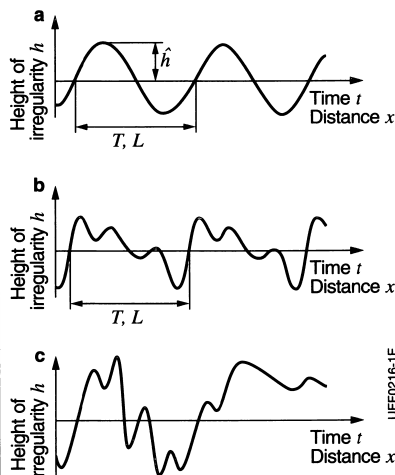
ε Phase shift,

$\Omega = 2\pi/L$ Breakdown angular frequency,

$\omega = v\Omega = 2\pi v/L$ Time angular frequency.

Figure 3: Types of courses of irregularity

- a) Sinusoidal course of irregularity,
b) Periodical course of irregularity,
c) Stochastic course of irregularity.
 T Period duration,
 L Period length,
 $\hat{h}$ Amplitude of irregularity height.



Periodical course of irregularity

In the next step, it is no longer only purely sinusoidal but also periodical courses of irregularity (Figure 3b) that are assumed, which means that expressing the relationship as a Fourier series (periodical functions can be described as an infinite series of sinusoidal oscillations) results in:

$$h(x) = h_0 + \sum_{k=1}^{\infty} \hat{h}_k \sin(\Omega x + \varepsilon_k) \quad (\text{Equation 3}),$$

$$h(t) = h_0 + \sum_{k=1}^{\infty} \hat{h}_k \sin(\omega t + \varepsilon_k) \quad (\text{Equation 4}),$$

where:

h_0 Base amplitude,

$\hat{h}_k$ Amplitude,

ε_k Phase shift,

$\Omega = 2\pi/L$ Breakdown angular frequency,

L Period length.

Stochastic course of irregularity

For real road surfaces, i.e. not only for periodical but also for random (stochastic) courses of irregularity (Figure 3c), in the transition into the complex syntax of the summation formula the integral is arrived at, thus moving from the discrete to the continuous spectrum [2]:

$$h(x) = \int_{-\infty}^{\infty} \hat{h}(\Omega) e^{j\Omega x} d\Omega \quad (\text{Equation 5}),$$

with the continuous amplitude spectrum:

$$\hat{h}(\Omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} h(x) e^{-j\Omega x} dx \quad (\text{Equation 6}),$$

or in the time range:

$$h(t) = \int_{-\infty}^{\infty} \hat{h}(\omega) e^{j\omega t} d\omega \quad (\text{Equation 7}),$$

where

$$\hat{h}(\omega) = \frac{1}{v} \hat{h}(\Omega) \quad (\text{Equation 8}).$$

Power spectral density of the road irregularities

As a rule, however, the course of irregularity as a function of path or time is less suitable for theoretical examinations of the road excitation. What is considerably more interesting is knowing the statistical mean of the excitations that occur when driving on a certain road surface, particularly in comparison with different road surfaces. This is why the power spectral

density $\Phi_h(\Omega)$ (PSD) depending on the breakdown angular frequency and power spectral density depending on the time angular frequency $\Phi_h(\omega)$ have prevailed as the measure for evaluation of road excitations. A mathematical definition and detailed discussion of these variables are not provided here. Refer to [1, 2]. For the following examination, the descriptive significance of the power spectral density that exists in the "energy in an infinitesimal frequency band" (with regard to path or time frequency) or the "power output in the natural frequency band" is sufficient.

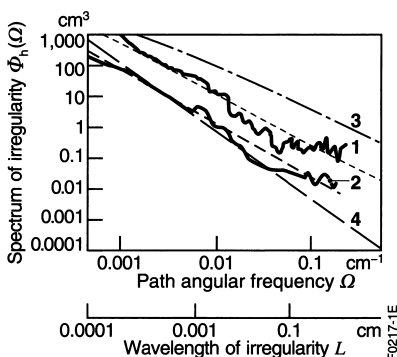
Taking account of $L = vT$, $\Phi_h(\Omega)$ and $\Phi_h(\omega)$ can be converted into one another. The following equation applies:

$$\Phi_h(\omega) = \frac{1}{v} \Phi_h(\Omega) \quad (\text{Equation 9}).$$

The power spectral densities specify the distribution of the power output of the excitation spectrum within the entire irregularity spectrum. This is usually represented in the double-logarithmic scale (Figure 4). An approximation of the courses of power spectral density on real road surfaces is possible here with straight lines, as described by the following equation:

Figure 4: Irregularity spectrum depending on the breakdown angular frequency

- 1 Highway,
- 2 Freeway,
- 3 Upper range limit for poor roads,
- 4 Lower range limit for good roads.



$$\Phi_h(\Omega) = \Phi_h(\Omega_0) \left(\frac{\Omega}{\Omega_0} \right)^{-w} \quad (\text{Equation 10})$$

where:
 $\Phi_h(\Omega_0)$ Power spectral density with a reference angular frequency Ω_0 (as a rule $\Omega_0 = 1 \text{ m}^{-1}$, which corresponds to a reference wavelength of $L_0 = 2\pi/\Omega_0 = 6.28 \text{ m}$),
 w Surface undulation of the road surface (from 1.7 to 3.3; for standard roads $w = 2$).

If the degree of irregularity and the surface undulation of a road surface are known, all the parameters are set in the above linear equation and it can then be used to characterize different types of road surface. The degree of irregularity is defined as the power spectral density at a reference angular frequency, i.e. $\Phi_h(\Omega_0)$.

Basic principles of the vibration characteristics of vehicles

The road excitations take effect on the vehicle body via the tires, wheel suspension, and the suspension / shock absorber

system. For theoretical analyses, vibration models of different complexity can be applied. With rising model complexity, there is an increase in the degrees of freedom and linked differential equations. To improve clarity, the fundamental relationships within vibrating vehicle systems will be explained on the basis of the dual-mass oscillator (Figure 5). More detailed information can be found in [1], [2], and [3].

On specification of the masses, the rigidities, and damping factors, all of the parameters have been set in the dual-mass model for technical analyses of the vibrations and two differential equations can be set up with the variables designated in Figure 5:

$$m_A \ddot{z}_A + k_{A,R} (\dot{z}_A - \dot{z}_R) + c_{A,R} (z_A - z_R) = 0 \quad (\text{Equation 11a}),$$

$$m_R \ddot{z}_R - k_{A,R} (\dot{z}_A - \dot{z}_R) - c_{A,R} (z_A - z_R) + c_R \dot{z}_R + k_R z_R = c_R h + k_R \dot{h} \quad (\text{Equation 11b}).$$

The division of Equation 11a and Equation 11b by the mass in each case leads to the normal form of a differential equation of the 2nd order and delivers the damped natural frequency ω_g and the undamped natural angular frequency ω_u as well as the damping factors for the wheel D_R and the body D_A .

The following applies to the uncoupled and undamped natural angular frequency at the wheel:

$$\omega_R = \sqrt{\frac{c_R + c_{A,R}}{m_R}} \approx \sqrt{\frac{c_R}{m_R}} \quad (\text{as } c_R \approx 10 c_{A,R}) \quad (\text{Equation 12}).$$

The following applies accordingly to the body:

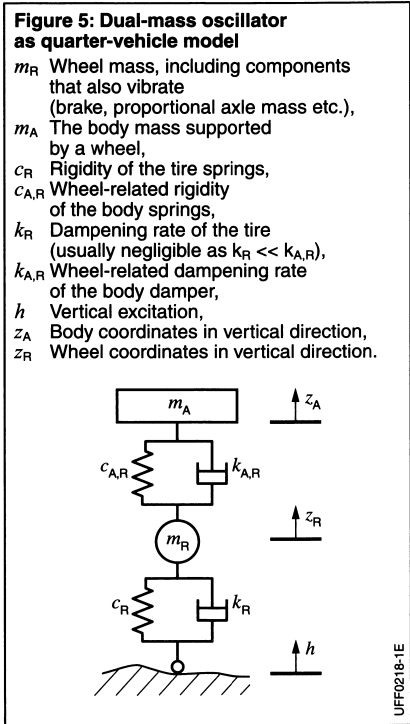
$$\omega_A = \sqrt{\frac{c_{A,R}}{m_A}} \quad (\text{Equation 13}).$$

In general, the damped natural angular frequency ω_g is calculated according to:

$$\omega_g = \omega_u \sqrt{1 - D^2} \quad (\text{Equation 14}),$$

whereby the following is also assumed as an approximation:

$$\omega_g \approx 0.9 \omega_u.$$



The following applies to the damping factor D_R at the wheel:

$$D_R = \frac{k_{A,R}}{2m_R\omega_R} = \frac{k_{A,R}}{2\sqrt{(c_{A,R} + c_{A,R})m_R}} = \frac{m_A\omega_A}{m_R\omega_R} D_A \quad (\text{Equation 15}),$$

whereby experience has shown that $D_R \approx 0.4$ is to be aimed for. The same applies to the body:

$$D_A = \frac{k_{A,R}}{2m_A\omega_A} = \frac{k_{A,R}}{2\sqrt{(c_{A,R}m_R)}} \quad (\text{Equation 16}).$$

In this case, $D_A \approx 0.3$ has proven to be effective. The dynamic wheel load fluctuations ΔG result in:

$$\Delta G = m_R\ddot{z}_R + m_A\ddot{z}_A = c_R(h - z_R) + k_R(\dot{h} - \dot{z}_R) \quad (\text{Equation 17}).$$

These variables together form a basis for the rough configuration of the suspension/shock absorber system of a motor vehicle.

If a body natural frequency (usually $f_A \approx 1$ Hz) is specified and the body mass (or the proportion of the body mass on a wheel) is known, the wheel-related body spring rigidity can be determined:

$$c_{A,R} = \omega_A^2 m_A \quad (\text{Equation 18}).$$

The conversion to the actual rigidity of the body springs takes place taking account of the ratio i between the wheel and spring movement in accordance with Figure 6. First of all, the actual spring force

$$\Delta F_F = c_A \Delta z_F \quad (\text{Equation 19})$$

and the wheel force ΔF_R are formulated with a spring compression Δz_R . The following applies to ΔF_R :

$$\Delta F_R = c_{A,R} \Delta z_R \quad (\text{Equation 20}).$$

The torque equilibrium around the pivot in Figure 6 leads to:

$$c_{A,R} \Delta z_R d_2 = c_A \Delta z_F (d_2 - d_1) \quad (\text{Equation 21}).$$

This can be used to convert the actual spring rigidity c_A in accordance with the geometric relationships to the wheel-related rigidity $c_{A,R}$:

$$c_{A,R} = c_A i^2 \quad (\text{Equation 22})$$

with the spring ratio i

$$i = \frac{(d_2 - d_1)}{d_2} = \frac{\Delta z_F}{\Delta z_R} \quad (\text{Equation 23}).$$

The same holds true for the vibration damper. For calculation of the body dampening rate (its effect on the wheel), the following applies in relation to the wheel in accordance with Equation 16:

$$k_{A,R} = 2D_A \sqrt{c_{A,R}m_R} \quad (\text{Equation 24}).$$

With $D_A = 0.3$ (see above) and m_A as a known variable of the vehicle under examination, the body dampening rate can be determined taking account of Equation 23.

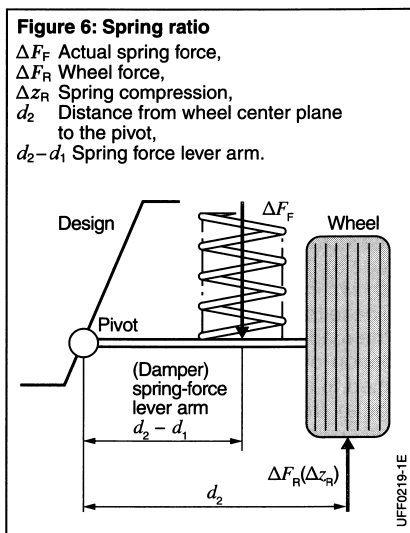


Table 1: Effects of changes to the suspension/shock absorber system on the vertical vehicle vibration characteristics

	Design parameters	Effects in the low-frequency range (body natural frequency)	Effects in the middle frequency range	Effects in the higher-frequency range (axle natural frequency)
Body data	Spring constant	Body natural frequency and body acceleration drop radically when the spring rigidity is reduced	Body acceleration drops slightly on reduction of the spring rigidity	Body acceleration remains virtually constant on reduction of the spring rigidity
	Dampening rate	Body acceleration rises radically on reduction of the dampening rate	Body acceleration drops radically on reduction of the dampening rate	Dynamic wheel load fluctuation rises radically on reduction of the dampening rate
Tire data	Spring constant	Natural frequency and amplitude remain approximately constant		Natural frequency and amplitude of the body acceleration and wheel load fluctuations drop approximately proportionally to the reduction in vertical wheel spring rigidity
	Dampening rate	Natural frequency and amplitude remain approximately constant		Amplitude of the body acceleration and wheel load fluctuation drops slightly with increasing damping at consistent wheel natural frequency

Using Equation 15, the estimate of optimal relationships between wheel and body mass with the help of the damping factors $D_R = 0.4$ and $D_A = 0.3$ to be aimed for delivers

$$D_R = \frac{m_A \omega_A}{m_R \omega_R} D_A$$

the following relationship:

$$\frac{m_A}{m_R} = \frac{0.4 \omega_R}{0.3 \omega_A} = \frac{0.4 f_R}{0.3 f_A} \quad (\text{Equation 25}),$$

whereby $f_R = \omega_R/2\pi$ and $f_A = \omega_A/2\pi$ were set. With $f_R = 12$ Hz and $f_A = 1$ Hz, the result is

$$m_R = \frac{1}{16} m_A .$$

The influence of different suspension/shock absorber parameters and their effects on different frequency ranges is shown in Table 1.

Transfer function and power spectral density of body acceleration and wheel load fluctuations

Equation 11a and Equation 11b can be sued to calculate the transfer functions (enlargement functions). The amount of the transfer function is the quotient from the stationary proportion of the amplitude of the output variables (body acceleration or wheel load fluctuation) and the amplitude of the input variable (vertical excitation, see Figure 5). The principle of the progression is shown in Figures 7c and 7d.

These transfer functions are now linked to the road irregularities. To do so, the distribution of the road irregularities is assumed to be a function of the breakdown angular frequency (Figure 8a) and the breakdown angular frequency is converted using the driving speed v (Figure 8b) into a time-dependent exciter angular frequency (Figure 8c). It can be seen from

Figure 8c that the exciter amplitudes at low frequencies are significantly greater than is the case at high frequencies. Transfer functions for the body acceleration (Figure 7c) and the wheel load fluctuation (Figure 7d) are squared and multiplied by this excitation spectrum. The results are the power spectral densities for the body acceleration (Figure 7a) and for the wheel load fluctuation (Figure 7b) [2].

The higher excitation amplitudes of the road surface at low frequencies raise the vibration amplitudes in the natural frequency range of the body (1 to 2 Hz). In the natural frequency range of the wheel (10 to 14 Hz), however, the low irregularity amplitudes lead to drops in the power spectral densities compared to the transfer functions, which means the body move-

ments become more dominant than the axle movements.

Quantification of vehicle vibrations

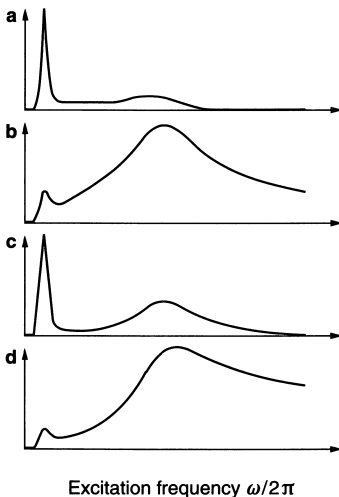
Only a short time after the introduction of the automobile, the first approaches to quantifying driving comfort in vehicles were developed at the start of the 20th century. The influence of comfort – particularly in the case of professional drivers – on fitness (i.e. ensuring good physical and mental condition) as well as on the health of the driver and thus on overall road safety was recognized very quickly. Accordingly, attempts were made to shape meaningful variables to describe comfort and to compare these in the form of objective comfort factors for different variants of vehicles and roads. Since then, a number of different assessment methods have been defined, based almost exclusively on acceleration signals at and around the driver's seat. Here, the frequency-dependent sensitivity of people to vibrations and oscillations initiated at different positions as well as different types and directions (translatory or rotatory; x , y , z direction) is taken into account with the help of the corresponding frequency weighting filters.

Two widely used approaches are the VDI guideline 2057 [4] and the standard

Figure 7: Transfer function and power spectral density for body acceleration and wheel load fluctuations

(According to [2])

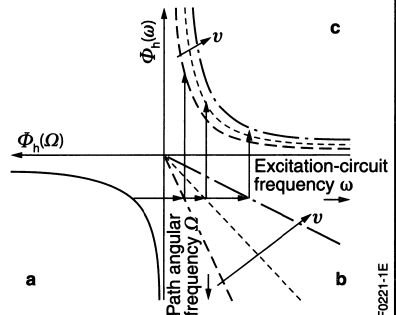
- a) Power spectral density, body acceleration,
- b) Power spectral density, wheel load fluctuation,
- c) Amount of transfer function, body acceleration,
- d) Amount of transfer function, wheel load fluctuation.



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Figure 8: Relationship between spectral densities dependent on time and breakdown angular frequencies

- a) Distribution of road irregularities as a function of breakdown angular frequency,
- b) Driving speed,
- c) Time-dependent exciter-circuit frequency.



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ISO 2631 [9]. Despite the relatively low overhead involved in these analyses compared to other methods, these have high informative value. At the same time, these form the basis for many approaches to comfort assessment that were developed in the 1990s and later (see [4], [5], [6], [7]). In the methods used to date, it is normally the case that the accelerations caused by the excitation at different measuring points (initiation positions) are used to determine an overall comfort factor (possibly from a number of partial comfort factors at different measuring points) that is to represent a measure of the load on the occupants. The accelerations are first of all frequency-weighted (i.e. amplitudes of different frequencies are linked to frequency-dependent factors) and subsequently reduced to a value, for example with the RMS method (Root Mean Square, see [4]) in accordance with:

$$\bar{a}_{\text{RMS}} = \sqrt{\frac{1}{T} \int_0^T a_w^2(t) dt} \quad (\text{Equation 26})$$

or with the RMQ method (Root Mean Quad, see [6]) in accordance with:

$$\bar{a}_{\text{RMQ}} = \sqrt[4]{\frac{1}{T} \int_0^T a_w^4(t) dt} \quad (\text{Equation 27}).$$

Depending on the number of evaluated measuring points, an overall factor has to be formed in a subsequent operation. For the VDI guideline 2057, only the evaluation of the measuring point with the highest load is relevant. All translatory accelerations of this initiation position (hand, seat or foot) are taken into account equally in the calculation of the factor. However, no uniform specifications for weighting individual initiation positions have yet been developed. The reason for this is that many of the general conditions of the subject tests used to determine the totaling rule for the overall comfort factor formed from the partial comfort factors deviated greatly from one another.

In order to ensure comparability of the factors of individual calculation methods, the overall comfort factor is frequently standardized to the standard VDI scale in a subsequent operation. At the same time, the objective factors are assigned verbalized levels of load intensity (as an aid to evaluation for the test subjects).

The differences between the individual comfort assessment methods (see [4], [5], [6], [7]) lie mainly in the number of measuring points to be evaluated, in the calculation specification for formation of the partial comfort factor from the temporal acceleration signal (RMS and RMQ method), and finally in the scattered weighting of different acceleration signals on merging to form an overall comfort factor. Over and above this, the question regarding the frequency weighting filter to be used depending on the measuring point has not been clarified conclusively. The frequency weighting functions, most of which have been determined empirically, deviate strongly from one another, particularly in low frequency ranges (below 1 Hz).

Experts also criticize the fact that the grouping of different acceleration types and directions to form a single comfort factor involves an information loss. This means that, under certain circumstances, effects such as a partial factor rise with a simultaneous drop in other partial factors at the same measuring point (for example of another direction) and the resulting compensatory influence on the overall comfort value remain undetected. This can be avoided by evaluating the temporal progressions of the frequency-evaluated accelerations immediately before formation of the partial comfort value.

Measures for vibration minimization

Road excitations lead to axle and body vibrations in vehicles. These are to be avoided in the course of minimizing dynamic wheel load fluctuations (safety) and body accelerations (comfort). They are reduced by means of the corresponding components within the wheel suspension, whereby different systems for suspension, damping, and suspension linkage have proven effective depending on the area of application.

Body springs and body dampers are traditionally used for these tasks. Coupled with kinematics and elastokinematics in the wheel suspension that are configured in line with requirements, these are intended to ensure optimal power transmission between the tires and road surfaces and, simultaneously, high levels of comfort.

However, with the increasing use of actuators in the area of chassis systems (for example superimposed steering or torque vectoring), the application overhead for optimization of the subsystems is becoming increasingly important. One of the most important reasons for this is the increase in the product added value due to low-cost function integration on the software side (i.e. the complete implementation of various functions of different subsystems in the software environment without being forced to increase the technological overhead). In addition, the technical and economic synergy potentials in overall vehicle networking can be increased significantly and the differentiation potential of various derivatives can be enhanced using the same electronic and software modules. This means that electronics and software are becoming increasingly important in the area of chassis systems, requiring interdisciplinary cooperation between hardware, electronics, and software developers at very early phases of development to enable the efficient and low-cost development of mechatronic systems (see [8]), particularly where safety is critical.

References

- [1] B. Heissing, M. Ersoy (Editors): *Fahrwerkhandbuch*. 1st Edition, Vieweg Verlag, 2007.
- [2] M. Mitschke, H. Wallentowitz: *Dynamik der Kraftfahrzeuge*. 4th Edition, Springer-Verlag 2004.
- [3] J. Reimpell, J.W. Betzler: *Fahrwerktechnik – Grundlagen*. 5th Edition, Vogel Verlag, 2005.
- [4] VDI guideline 2057: Human exposure to mechanical vibration, whole-body vibration. VDI-Gesellschaft Entwicklung Konstruktion Vertrieb, Düsseldorf 2002.
- [5] S. Cucuz: *Auswirkung von stochastischen Unebenheiten und Einzelhindernissen der realen Fahrbahn*. Dissertation, University of Braunschweig, 1992.
- [6] D. Hennecke: *Zur Bewertung des Schwingungskomforts von Pkw bei instationären Anregungen*. VDI Progress Reports, Düsseldorf, 1995.
- [7] M. Mitschke, S. Cucuz, D. Hennecke: *Bewertung und Summationsmechanismen von ungleichmäßig regellosen Schwingungen*. ATZ 97/11 (1995).
- [8] ISO 26262: 2011: Road vehicles – Functional safety.
- [9] ISO 2631 AMD1 (2010): Mechanical vibration and shock – Evaluation of human exposure to whole-body vibration.

Suspension

Basic principles

The suspension system of a vehicle has a decisive influence on the vibration characteristics and therefore on both comfort and driving safety. Depending on the vehicle category and use case, different solutions have prevailed in the meantime. An overview of the different suspension design elements is shown in Figure 1, using a quarter vehicle as an example.

As a general principle, suspension design elements include all parts of the wheel suspension of a motor vehicle that deliver return forces in the case of elastic deformation. The media that perform the suspension work on the different suspension systems are either steel (spring steel), polymer materials (rubber), or a gas (air).

Tires

As the connecting element between the road surface and vehicle, the tire is the first suspension design element in the transfer chain from excitation to the occupants that has a decisive influence on both the comfort (acoustics, rolling characteristics) and driving safety (longitudinal and lateral force potential). It influences both suspen-

sion and damping characteristics, whereby these are not sufficient to eliminate the need for other vibration-absorbing elements in modern vehicles. An exception here is for example construction machines, where changed comfort requirements mean that in almost all cases the suspension and damping are achieved by means of the tires.

Elastomer mounts

Elastomer mounts are rubber elements with different functions and properties that interconnect individual components of a chassis system or secure them to the body.

The rubber mounts are used to provide insulation against vibrations and thus enhance comfort, particularly in the case of higher-frequency excitation (acoustics). At the same time, a suitable configuration (geometry, rigidity of the rubber mounts etc.) can decisively influence the driving dynamics.

In contrast to series-production vehicles, uniball joints (not rubber-mounted connections between wheel suspension and body) are deployed in motor racing; this improves driving dynamics to the detriment of comfort.

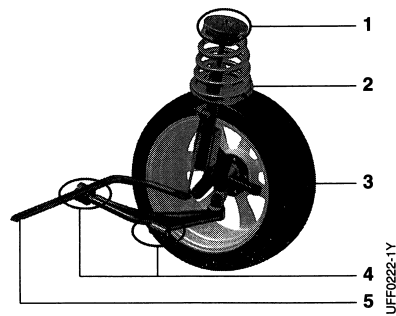
In order to reconcile the conflicting objectives of soft mounts for high comfort and rigid mounts for high driving dynamics, in the meantime increasing use is made of adaptive or active suspension mounts that are able to adapt their properties to each driving situation.

Body springs

Body springs are parts of the chassis system that provide most of the vertical return forces between the wheel and body. Depending on the use case, various types of spring with very varied properties are used. A summary of the characteristics of suspension design elements used in vehicle construction can be found in Table 1.

Figure 1: Suspension design elements within the wheel suspension (McPherson split axle as an example)

- 1 Dome mount (rubber mount),
- 2 Body spring,
- 3 Tires,
- 4 Rubber mounts,
- 5 Stabilizer.



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Table 1: Suspension design elements in vehicle construction

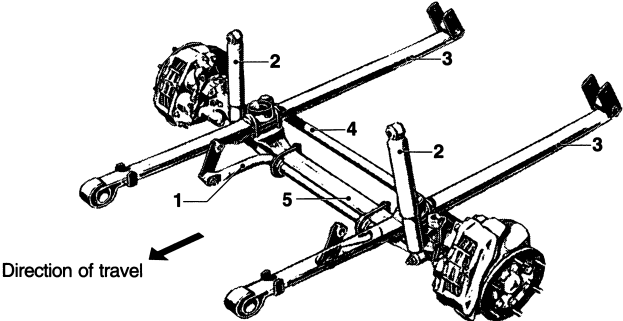
Suspension design element	Load influence on body natural frequency	Advantage	Disadvantage	Properties
Steel springs				
Leaf spring	– Natural frequency drops with increasing load – Characteristic curves are generally linear	– Good transfer of forces to chassis (trucks) – Low cost	– Maintenance requirement – Friction damping usually inadequate – Acoustic influences	– Assumption of the suspension linkage function is possible – Single-layer or multilayer version – Depending on type, subject to friction (in the passenger car, an intermediate plastic layer reduces friction → positive influence on acoustics)
Coil spring		– Great leeway for configuration – Low cost – No intrinsic damping – Lower space requirement – Low weight – Maintenance-free	– Separate elements required for suspension linkage – Spring characteristic curve is non-variable	– Arrangement of the shock absorbers within the springs is possible – Progressive characteristic curve due to corresponding geometry of the springs can be implemented (variable upward gradient or conical wire)
Torsion-bar spring		– Wear-free and maintenance-free – Depending on design, ride height adjustment also possible	– Long springs – Wheel-related spring rigidity depends on suspension arm arrangement	– Made of round steel (for low weight) or flat steel (with increased stress)
Stabilizer	– No influence with suspension on the same side – Half of stabilizer rigidity effective with one-sided suspension – Entire stabilizer rigidity effective with alternating suspension	– Simple possibility to influence driving dynamics of a vehicle – Reduction in the roll angle – With deployment of active systems improved comfort and enhanced driving dynamics	– Additional weight – Costs	– Influence on cornering characteristics (oversteer or understeer) – U-shaped, bent fully round or tube material is usual – Stub often flat-rolled due to bending stress – Stabilizer attachment points located far out on the axle to achieve small diameter – Axes of rotation of suspension arms configured so that stabilizer load is only torsion (not flexion)

Table 1: Suspension design elements in vehicle construction (continued)

Suspension design element	Load influence on body natural frequency	Advantage	Disadvantage	Properties
Air springs and hydropneumatic springs				
U-type bellows gas springs and air springs with bellows	– Natural frequency remains constant with increasing load – Characteristic curves depend on gas properties, form of rolling piston shape, and cord angle in the bellows	– Comfort characteristics independent of payload	– Separate elements required for suspension linkage – Low pressure (< 10 bar) requires high volumes	– Implementation of soft, vertical spring rigidity – As spring strut or individual spring, can be found above all in commercial vehicles and buses – Increasingly, deployment in passenger cars for ride height control of the rear axle and all-round suspension
Hydro-pneumatic spring	– Natural frequency rises with increasing load due to nonlinear spring rigidity	– Hydraulic damping and ride height control easy to integrate	– Maintenance requirement for rubber membrane due to diffusion tendency	– Gas volume in the spring accumulator determines the suspension characteristics – Power flux by means of gas and oil – Integration of the damper valves in the shock absorber and in the connection between the suspension strut and accumulator
Rubber springs				
Rubber spring	– Natural frequency is influenced by increased load due to the nonlinear spring rigidity	– Design is very freely definable – Low cost	– Limited temperature range – Aging	– Vulcanized rubber shear spring between metal parts – Used as assembly mountings (engine and transmission), suspension-arm mountings, additional springs etc. – Increasingly with integrated hydraulic damping

Figure 2: Example of a leaf spring with suspension linkage function

1 Stabilizer, 2 Damper, 3 Leaf springs, 4 Panhard rod, 5 Rigid axle.



Types of spring

Leaf springs

The oldest types of spring used in vehicle construction are leaf springs, which were even used on horse-drawn coaches (Figure 2). Alongside the suspension functionality, major advantage of this type of spring is their possible use as a suspension linkage engineering design element to connect the body and axle. Multilayer leaf springs also have damping characteristics which, however, can lead to negative responses (acoustic influences). The damping forces that can be achieved are not sufficient to completely eliminate the need for conventional shock absorbers.

The influences on comfort and their weight mean that leaf springs in the meantime no longer meet market requirements for personal transportation and they are therefore only used now in a few passenger cars (light utility vans, off-road vehicles). It is still usual to deploy this type of spring in the area of commercial vehicles on account of the low costs and high reliability.

Helical springs

The great leeway for configuration with simultaneously low costs means that helical springs are the most frequently deployed type of body springs in the field of passenger cars. On this type of spring, the return forces are generated by the elastic torsion of individual coils during the change in length.

As helical springs are mainly able to absorb forces in the direction of the longitudinal axis of the spring, when they are used as body springs the other force components are braced by the suspension linkage.

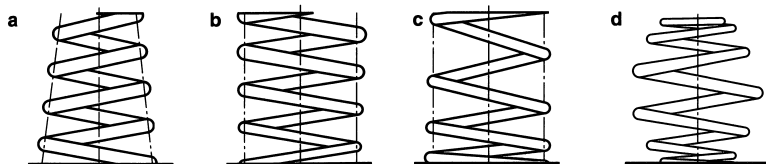
A geometric configuration of the springs that is in line with requirements (wire thickness, coil diameter and spacing, Figure 3) means that not only different design envelopes but also different spring rigidity progressions over the compression path can be achieved. This can be used in turn to influence the load-dependent body natural frequency and thus the driving comfort.

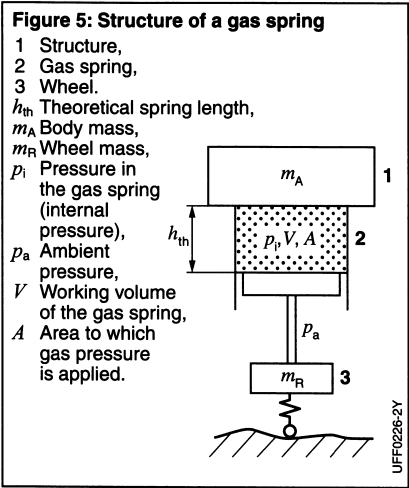
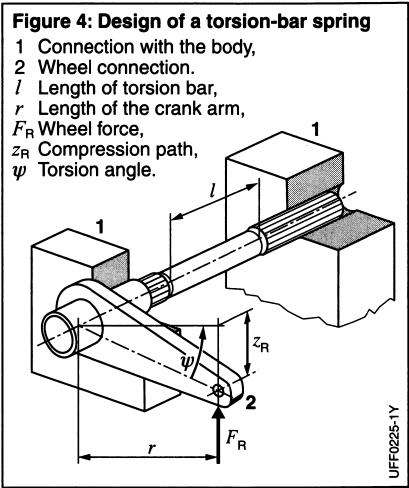
Torsion-bar springs

This type of spring is encountered mainly in passenger cars and light utility vans. They are bars made of spring steel that are subjected to torsion. The fixed clamping of one end of the bar and the rotatable mounting of the other end means that the shaft is elastically torsioned when subjected to a load in the form of torque applied in the direction of its axis. In the motor vehicle, the elastic twisting of the torsion bar is achieved with the help of a crank secured to the rotatable end of the rod (Figure 4). As a rule, the crank arms are suspension arms of the axle or wheel suspension. The torsion-bar springs are usually arranged in the bearing center of the suspension arms on the body side on the opposite end of which the vertical wheel force F_R takes effect as the external load.

Figure 3: Design examples of different types of helical springs

- a) Changeable coil diameter,
- b) Changeable wire diameter,
- c) Changeable coil spacing,
- d) Mini-block spring (combination of a, b, and c).





Gas springs

The body springs presented so far are fixed, springing media, whereby the work was performed by the change in shape of the steel springs. In contrast to this, the spring work in the case of gas springs is provided by a change in volume of the gas. The body of the vehicle is decoupled by an effective gas volume (possibly also by an additional fluid, see hydropneumatic springs) from the excitation and it vibrates on the gas cushion within the gas spring (Figure 5). This results in a favorable possibility to integrate a ride height control function that can be implemented by pumping the intermediate medium (gas or fluid) in or out.

A characteristic parameter of the gas spring is the "theoretical spring length" h_{th} , which results as a quotient from the compression-dependent working volume $V(z)$ (including any additional volume) and the effective surface area A to which gas pressure is applied:

$$h_{th} = \frac{V(z)}{A} \tag{Equation 1}$$

where:
 z Compression path.

The equation for the spring force F :

$$F = (p_i - p_a) A \tag{Equation 2}$$

where
 p_a Ambient pressure,
 p_i Internal pressure,

generally leads to the following for the spring rigidity of a gas spring [1]:

$$c(z) = A n p(z) \frac{1}{h_{th}} \tag{Equation 3}.$$

The polytropic exponent for isothermic and slow spring movements is $n = 1$ – for adiabatic and faster spring movements it is $n = 1.4$. The natural angular frequencies are calculated in the same way as for steel springs:

$$\omega_{Gas} = \sqrt{\frac{c}{m}} = \sqrt{\frac{c(z)g}{(p - p_a)A}} = \sqrt{\frac{g n p(z)}{(p - p_a)h_{th}}} \tag{Equation 4}.$$

When the requirement for a relatively small spring diameter is met $p_i \gg p_a$. This simplifies the equation for the natural angular frequency to:

$$\omega_{Gas} = \sqrt{\frac{g n}{h_{th}}} \tag{Equation 5}.$$

However, the above-mentioned theoretical piston cylinder gas spring is only used in vehicles in a modified form, whereby in principle a distinction is made between two types of gas spring, the bellows air spring and the hydropneumatic spring. The fundamental difference with regard to the vertical dynamics lies in the influence of the load on driving smoothness and in the different effects on the spring rigidity in the level balancing of both systems. Whereas in the case of the hydropneumatic type level balancing is achieved by pumping in fluid (with constant mass of the gas in the spring), level balancing on air springs with bellows takes place by pumping a gas (air) into the spring, thus restoring the original suspension volume. The change in spring rigidity of the hydropneumatic spring type that this causes also leads to load dependence of the body natural frequency. In contrast to this, the air spring with bellows has a virtually constant body natural frequency in the entire load range.

To conclude, Figure 6 shows the influence of different suspension systems on the natural frequency, and thus also indirectly on the comfort with rising loads. The reason for the influence of the body natural frequency on comfort lies in the different

resonance ranges of different organs in the human body and the consequence that an excitation of human body parts with their natural frequency impairs well-being. This is why a body natural frequency, below the resonant frequencies of the human body, that is as independent as possible of the load is to be ensured.

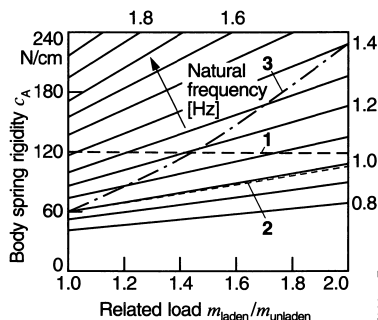
However, Figure 6 also clearly indicates that an approximately constant natural frequency with rising load is only present in the case of air springs. With steel springs, the natural frequency drops due to the constant spring rigidity; with the hydropneumatic system, on the other hand, it increases.

Air springs with bellows

Gas springs with bellows with pneumatic ride height control are suspension systems with constant gas volume (see above), whereby these are in turn divided into two categories. These are firstly gas springs with bellows and, secondly, U-type bellows gas springs (Figure 7) which, in a similar way to pneumatic tires, consist of rubber material reinforced by woven textiles. On

Figure 6: Comparison of different suspension systems depending on payload

- 1 Steel spring,
- 2 Air spring,
- 3 Hydropneumatic spring.

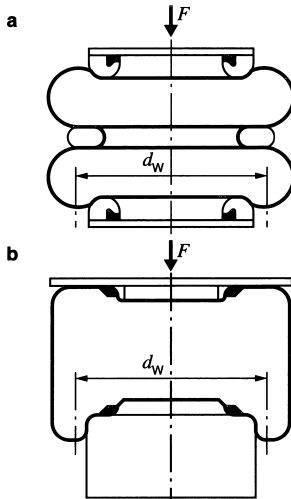


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Figure 7: Gas springs with bellows

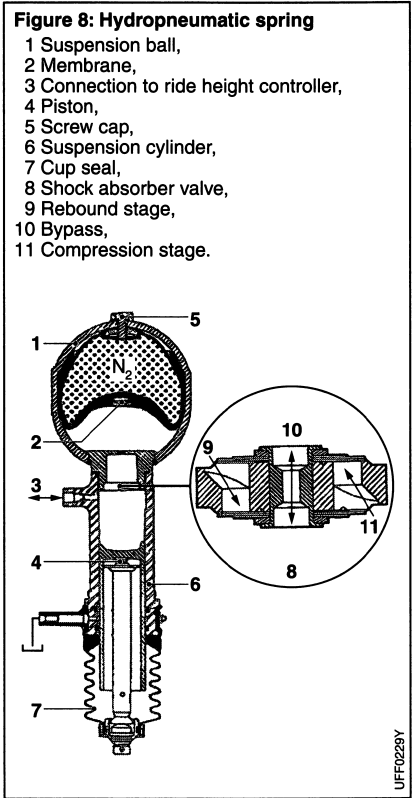
- a) Bellows,
- b) U-type bellows.

F Force,
 d_w Effective diameter of the air springs.



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these systems, the ride height control is implemented by pumping gas into or releasing gas from the spring with the suspension volume remaining constant as a general principle. The effective surface area of the air springs (and thus the gradient of the return force) that is affected by the overpressure is not usually constant, rather it changes throughout the stroke. This enables specific influence on the load-bearing capacity by designing the contours of the rolling piston of the U-type bellows gas springs (and thus a change in the effective surface area A in Equation 3 via the range of the spring) accordingly. The effective surface area A of the air spring can be determined via the effective diameter. With integration of an additional volume (increase of h_{th} , see Figure 5), a less progressive characteristic curve can be achieved.



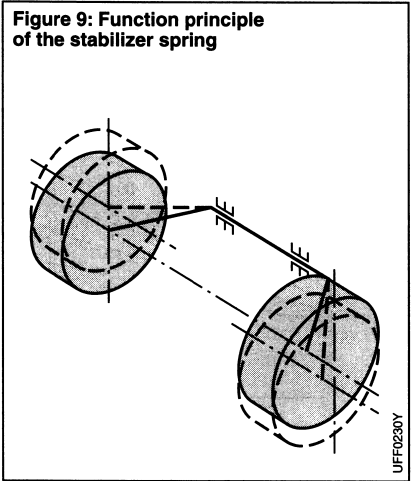
Hydropneumatic springs

In accordance with the considerations above, hydropneumatic springs (Figure 8) with integrated ride height control are gas springs with constant gas weight (see above), whereby the power flux is routed not only through a gas but additionally through a fluid. Here, the fluid and gas are separated by an impermeable rubber membrane. It is only when the fluid is placed in between that a wear-resistant and low-friction seal is achieved between the piston and cylinder.

Another advantage of this system lies in the possibility to integrate hydraulic damping in the suspension design element. A disadvantage, on the other hand, is the dependency of the natural frequency on the load (influence on comfort). The reason for this is the pumping in and out of the fluids at a constant gas weight that is required for ride height control. The load-sensitive volume change of the gas leads to a shift in the spring rigidity in such a way that with rising load there is a fundamental increase in natural frequency (see Figure 6).

Stabilizers

The suspension systems described above are deployed primarily for the vertical suspension of a vehicle. For the roll suspension, on the other hand, additional passive or active stabilizer springs (under certain circumstances with additional roll damping)



are used alongside classical body springs. A diagram of the principle is provided in Figure 9. In the event of a rolling movement of the body, i.e. spring compression of the wheels of one axle in opposite directions, the stabilizer is torsioned and delivers an aligning torque around the roll axis. In the case of purely vertical motions of one axle, on the other hand, this has no effect. If the proportions of the rolling moment braced by the stabilizers on the front and rear axles have a different ratio to the proportions braced by the body springs, not only is the roll angle reduced but the breakdown of the differences in wheel load of an axle on cornering is also influenced.

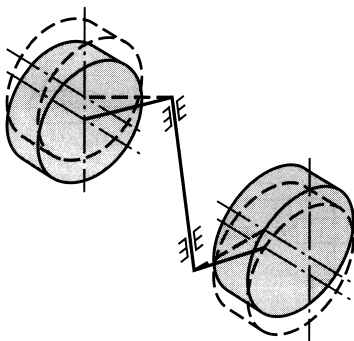
On a vehicle with a corresponding stabilizer configuration, this can shift the driving characteristics towards understeer (increase in the roll rigidity on the front axle or reduction in the roll rigidity on the rear axle) or oversteer with all other parameters unchanged. In the case of active stabilizers, the stabilizer force can also be actively influenced and adapted to suit the driving status. This enables a reduction in for example copy effects on one axle (due to decoupling between the right-hand and left-hand sides) when the vehicle is driven straight ahead but it also increases the driving dynamics on cornering by minimizing the inclination of the body. Here, the stabilizers do not influence the vertical vibration characteristics of the vehicle.

Compensating springs

Compensating springs have the opposite effect to that of the stabilizer (Figure 10). As a pure stroke element, they have no effect during rolling motions of the body.

The compensating spring was used in the past on axle designs where the wheel suspension kinematics required wheel load differences that were as low as possible to suppress the "resting effect" (stronger rebound of the wheel on the inside of the curve compared to the spring compression of the wheel on the outside of the curve when cornering). The rigidity of the body springs and this braced proportion of the rolling moment on the axle under examination could then be reduced accordingly. Compensating springs are no longer in modern passenger car wheel suspensions.

Figure 10: Function principle of the compensating spring



Suspension systems

Increasing customer requirements (comfort and driving dynamics) with regard to passenger cars and the strongly fluctuating load states of commercial vehicles mean that exclusive deployment of conventional steel springs is often insufficient. In such cases, either partially loaded or fully loaded suspension systems are used.

The integration of the additional functions of partially loaded or fully loaded systems enables increases in both comfort and driving dynamics (for example transverse locking of the springs of one axle to enhance stability on cornering).

Partially loaded systems

These systems are characterized by the fact that the forces to be braced by the suspension system are divided between steel and air springs according to a specified ratio.

In the case of soft body springs (to enhance driving comfort), wide spring ranges occur for example when a vehicle is loaded. In order to prevent the vehicle body from being lowered too much, ride height control systems with air springs or hydropneumatic springs are used. Here, sensors determine

the ride height and provide this information to the control system. By pumping in or releasing air (air springs with bellows) or oil (hydropneumatic springs), the ride height can be then adapted in line with requirements. Depending on the vehicle segment (passenger cars or commercial vehicles), ride height control systems offer different additional functions.

In the passenger car, for example, a speed-dependent ride height control of the body in order to save fuel is possible. The adjustable ride height can also be used on poor-quality road surfaces to enhance the vehicle's capability to handle rough terrain.

In the case of commercial vehicles, on the other hand, ride height control enables variable adaptation of the loading area to different loading ramps. Other functions can also be implemented by networking with other systems. These include, for example, an automatic ride height increase if the lifting axle is raised, lowering on exceeding the maximum axle load, or brief raising of the lifting axle to increase the wheel load on the driven axle.

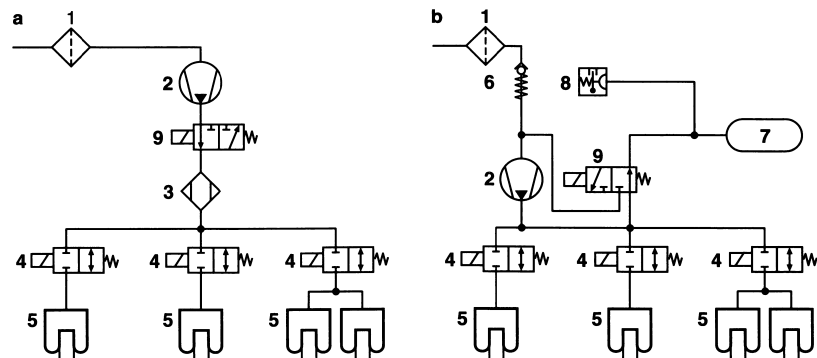
Figure 11: System architectures of ride height control, fully loaded systems

a) Open system, b) Closed system.

1 Filter, 2 Compressor, 3 Drier, 4 2/2-directional-control valve,

5 Bellows, 6 Non-return valve,

7 Pressure container, 8 Pressure switch, 9 3/2-directional-control valve.



Fully loaded suspension systems

In the case of fully loaded systems, as opposed to partially loaded solutions, the task of suspension is assumed solely by the gas springs and the helical springs are completely eliminated. Depending on the available hardware and control strategy, the ride height control can be on selected axles or on all axles. Embedding the control system architecture in the global vehicle control architecture ensures that a negative mutual influence of the axles is excluded, thus preventing, for example, the vehicle body from slanting.

As a general principle, the ride height control for fully loaded systems can be designed in the form of an open or closed system.

In the case of the open system, a compressor draws air from the atmosphere and provides it in compressed form to the air springs if required. The increase in pressure raises the vehicle body. To lower the ride height, air is expelled into the environment, thus lowering the pressure in the springs again. Although this system involves relatively low overhead in construction and has a simple control system, it has the decisive disadvantage that high compressor output has to be provided for the short periods of control operation. Moreover, an air drier is required and an acoustic load can be expected when air is drawn in and expelled.

The closed system draws air from a pressure accumulator of the suspension system and feeds this directly to the air springs. When the ride height is lowered, the compressed air is returned to the pressure tank. Although the compressor output requirements for this system are lower and the air drier can be eliminated because the working medium is already dry, other components (accumulator, pressure switch, non-return valve, return line etc.) are required, fundamentally increasing the overhead in construction compared to an open system.

The system architectures of the open and closed ride height control systems are compared in Figure 11. This clearly illustrates the greater complexity of the closed system.

Chassis systems

Compared to passive systems, active vehicle chassis systems enable optimal adaptation of spring and damper forces to all driving states and road irregularities. With the aid of external sources of energy, forces are generated that stabilize both the axles and the vehicle body. In the meantime, a number of different system architectures have been developed. These differ above all with regard to the overhead involved (design envelope and costs), the energy requirement, and the quality of control operations. A few of these systems are presented briefly below.

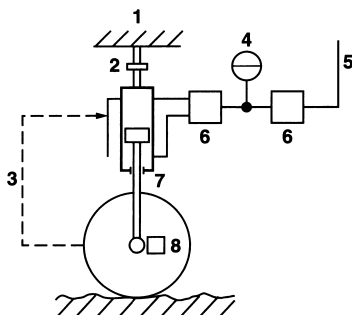
Systems with hydraulic cylinders

With this version, the body motion and body position are regulated with quickly adjustable hydraulic cylinders (Figure 12), whereby different items of sensor information (wheel load, range of spring, acceleration etc.) are used as input variables for the control operation.

The control system achieves virtually constant wheel loads while maintaining a constant mean ride height. The static wheel load in this case is carried by steel springs or hydropneumatic springs.

Figure 12: Active chassis system with hydraulic cylinder

- 1 Vehicle body,
- 2 Wheel-load sensor,
- 3 Travel sensor,
- 4 Accumulator,
- 5 Pump circuit,
- 6 Servo valve,
- 7 Positioning cylinder,
- 8 Acceleration sensor.



Systems with hydropneumatic suspension system

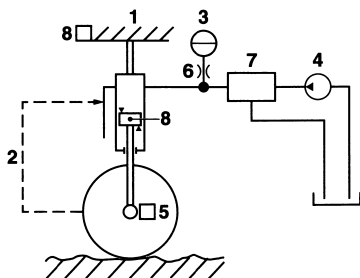
To stabilize the vehicle, a hydropneumatic suspension system uses specific oil regulation. This is done by pumping hydraulic oil into the suspension struts or draining it off the suspension struts (Figure 13). To limit the energy absorption, the control strategy of this system consists of control operations to eliminate longwave irregularities (low-frequency excitations). In the case of higher-frequency proportions, a gas volume takes effect in the proximity of the suspension strut. Here, the shock absorber is essentially geared to the wheel movements.

Version with spring-mounted point adjustment

In this system, the vehicle body is held horizontally in the low-frequency range in that a conventional helical spring is designed as variable at its mounting point (either against the vehicle body or in relation to the axle, Figure 14). The base is raised for spring compression and lowered for spring rebound. Control operations are continuous by means of a fluid pump and proportioning valves. The helical spring, however, must be longer compared to its original design.

Figure 13: Hydropneumatic system

- 1 Vehicle body,
- 2 Travel sensor,
- 3 Accumulator,
- 4 Pump circuit,
- 5 Acceleration sensor,
- 6 Throttle,
- 7 Proportioning valve,
- 8 Shock-absorber piston fitted with valves.



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In this type of system, the shock absorbers can be fitted with constant adjustment parameters and, above all, they can be matched to the wheel dampers.

Version with an air suspension system

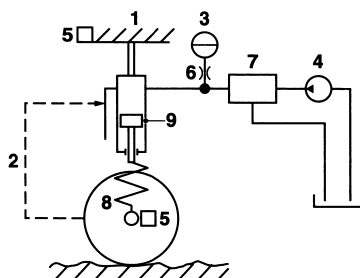
For high dynamics of an active air suspension system, rapid pressure changes (buildup and reduction) are required in the wheel-mounted air-spring bellows. This can be implemented by means of coupled shift elements (compressor output alone is not sufficient here), which, depending on the driving state, takes air out of the air spring on the inside of the curve and feeds it into the bellows on the outside of the curve (Figure 15) to keep the body horizontal. The drive of the shift unit can be either electric, hydraulic or a combination of both. Links to other vehicle systems are also possible.

Electromagnetic system

In the case of an electromagnetic active chassis system, linear electromagnetic motors are fitted on each wheel. These are able to actively compensate for the road irregularities. The linear motors are supplied with electrical energy by power amplifiers, whereby as a general principle force control or path control can be imple-

Figure 14: Spring-mounted point adjustment

- 1 Vehicle body,
- 2 Travel sensor,
- 3 Accumulator,
- 4 Pump circuit (oil),
- 5 Acceleration sensor,
- 6 Throttle,
- 7 Proportioning valve,
- 8 Body spring (helical spring),
- 9 Spring-mounted point adjuster.



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mented. Cross-axis system networking also enables compensation of rolling and pitching vibrations. In the solution that is currently implemented, the static wheel loads are absorbed via torsion springs at the wheels to limit the electrical energy requirement. A passive shock absorber is also used.

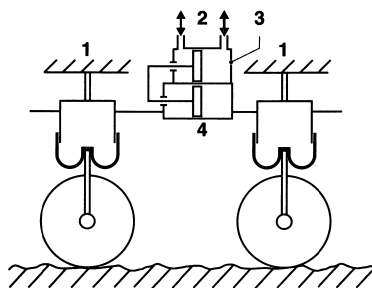
The advantages of the electromagnetic solution lie above all in the high adjustment speed. Despite the low outer dimensions, the electric motors have sufficient power output to ensure driving safety in all driving situations. In comparison with conventional systems, the disadvantages lie in the greater weight and in the increased costs, as an additional, non-electronic damping system is required for safety reasons. As a general principle, however, the linear electromagnetic motor can also be operated as an alternator, which means that energy can be recuperated. This enables a lowering of the power requirement of the overall system, which is specified for standard road surfaces as less than 1 kW.

References

- [1] B. Heissing, M. Ersoy (Editors): *Fahrwerkhandbuch*. 1st Edition, Vieweg Verlag, 2007.
- [2] M. Mitschke, H. Wallentowitz: *Dynamik der Kraftfahrzeuge*. 4th Edition, Springer-Verlag 2004.
- [3] J. Reimpell, J.W. Betzler: *Fahrwerktechnik – Grundlagen*. 5th Edition, Vogel Verlag, 2005.
- [4] L. Eckstein: *Vertikal- und Querdynamik von Kraftfahrzeugen*. ika/fka 2010.

Figure 15: Active system with pneumatic suspension

- 1 Vehicle body,
- 2 Auxiliary power,
- 3 Shift unit,
- 4 Air volume shift unit.



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Shock absorbers and vibration absorbers

Shock absorbers

The masses of the vehicle body and wheels connected by the body springs form a system that can vibrate and is excited by irregularities on the roadway and the dynamic movements of the vehicle. Shock absorbers are required to dampen the vibrating system. Nowadays, it is almost exclusively hydraulic telescopic shock absorbers that are used in motor vehicles as body shock absorbers; these convert the kinetic energy of the body and wheel vibrations into heat. The shock absorbers are configured taking account of the partially conflicting requirements for comfort (minimization of body accelerations) and driving safety (minimization of wheel load fluctuations).

Basic principles of hydraulic telescopic shock absorbers

The damper effect of hydraulic telescopic shock absorbers (Figure 1) is based on the movement subject to flow resistance of a shock absorber piston equipped with throttle elements (damper valves) within an oil-filled working cylinder. In the process, mechanical work is converted into heat that is released into the environment via the shock absorber surface. The pressure difference Δp between the two working chambers and the effective surfaces caused by the throttle elements on both sides of the shock absorber piston generates the resulting shock-absorber force F_D when the shock absorber moves in or out. The area on which the pressure prevailing in each working chamber is applied corresponds to an annular surface A_{KR} for the working chamber through which the piston rod of the shock absorber runs (see working chamber 1 in Figure 1). The outer diameter of the annular surface corresponds to the diameter D of the shock absorber piston; the inner diameter corresponds to the diameter d of the piston rod.

The following applies:

$$A_{KR} = \frac{\pi}{4} (D^2 - d^2).$$

In the other working chamber (see working chamber 2 in Figure 1), the effective area corresponds to the piston surface A_K which results from the diameter D of the shock absorber piston.

$$A_K = \frac{\pi}{4} D^2.$$

When there is a movement of the shock absorber piston (i.e. an inward or outward movement of the shock absorber), the changes to the volumes in the two working chambers lead to a flowing movement of the incompressible damping fluid between the working chambers of the shock absorber (in the case of twin-tube shock absorbers, additionally between a working chamber and the compensating chamber). The oil volume flows in each case must pass the corresponding shock absorber valves.

The oil volume flows through the relevant shock absorber valves result from the geometry of the shock absorber and the speed of inward or outward movement $\dot{z}$. The following applies to the volume flow $\dot{Q}_1$ between the two working chambers:

$$\dot{Q}_1 = \frac{\pi}{4} (D^2 - d^2) \dot{z}.$$

The inward or outward movement of the piston rod on hydraulic telescopic shock absorbers leads to a variable total volume of the working chambers that depends on the retracting or extending state. The incompressibility of the shock absorber oil means that a possibility to compensate for the oil volume displaced or released by the piston rod is required. The following applies to the volume flow $\dot{Q}_2$ of this compensating volume:

$$\dot{Q}_2 = \frac{\pi}{4} d^2 \dot{z}.$$

The volume flow $\dot{Q}$ through a shock absorber valve is linked via the through-flow characteristics of each valve to the prevailing pressure difference Δp . The

throughflow characteristics of a valve result from the joint effect of the throttle geometry (for example bore diameter of the flow-through channel) and any spring load (i.e. pressure-dependent variation of the discharge opening, Figure 1). The throughflow characteristics can be adapted to the needs of each situation by configuring and coordinating these parameters. The characteristics of the valves are to be designed in such a way that no cavitation whatsoever occurs (formation and implosion of gas bubbles in the working medium due to static pressure fluctuations in the range of the vapor pressure of the working medium) inside the shock absorber. Cavitation leads to acoustic problems and also to damage – ultimately to failure of a shock absorber.

Types of hydraulic telescopic shock absorbers

Single-tube shock absorbers

To balance out the retracting or extending piston rod volume, single-tube shock absorbers have an enclosed gas volume which is separated from the working chambers filled with shock absorber oil with the help of a moving dividing piston (Figure 1a). In the compression phase (moving in) of the shock absorber, the gas volume is compressed according to the volume flow $\dot{Q}_2$; in the rebound phase (moving out), it is relaxed according to the volume flow $\dot{Q}_2$. As a rule, the pressure of the gas volume is between 25 and 35 bar, which means that the maximum occurring retraction forces can be absorbed. The working volume flow $\dot{Q}_1$ flows through each of the corresponding shock absorber valves in the shock absorber piston. On moving in, it flows through the compression stage valve and on moving out through the rebound stage valve.

The high gas pressure means that the tendency for cavitation to occur is low in the case of the single-tube shock absorber. The heat that is generated can be released into the environment directly via the outer surface of the working cylinder. The advantages of lean design, the low weight and discretionary installation position of the single-tube shock absorber are offset by the great length, increased friction, and high requirements with re-

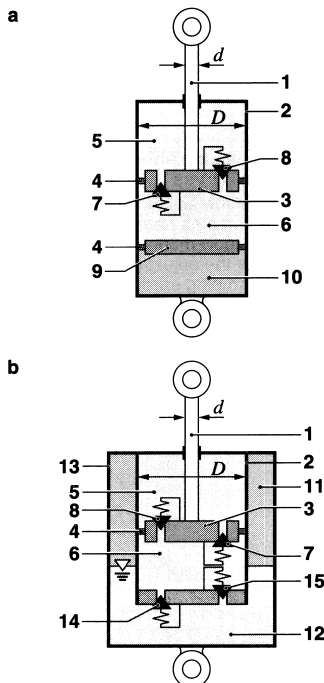
gard to sealing the piston rod and gas volume.

Twin-tube shock absorbers

Twin-tube shock absorbers have a compensating chamber resulting from the arrangement of an outer tube around the

Figure 1: Structure of single-tube and twin-tube shock absorbers

- a) Single-tube shock absorber,
- b) Twin-tube shock absorber.
- 1 Piston rod, 2 Working cylinder,
- 3 Shock absorber piston, 4 Piston seal,
- 5 Working chamber 1,
- 6 Working chamber 2,
- 7 Shock absorber valve (rebound stage valve),
- 8 Shock absorber valve (compression stage valve),
- 9 Dividing piston, 10 Gas volume,
- 11 Compensating chamber,
- 12 Reserve volume,
- 13 Outer tube,
- 14 Bottom valve (compression stage valve),
- 15 Bottom valve (rebound stage valve).
- D Inner diameter of working cylinder and diameter of shock absorber piston,
- d Diameter of piston rod.



working cylinder (Figure 1b). The compensating chamber balances out the retracting or extending piston rod volume. To achieve this, it is connected via bottom valves to the lower working chamber of the shock absorber. The compensating chamber is partly filled with shock absorber oil and partly with a gas (as a rule, air). The gas volume is usually at atmospheric pressure or slight overpressure (6 to 8 bar). The piston and bottom valves must be coordinated in such a way that no cavitation occurs. In the compression stage, i.e. when the shock absorber is moving in, the damping work is therefore done at the corresponding bottom valve through which the volume flow $\dot{Q}_2$ flows. On the other hand, the oil volume flow $\dot{Q}_1$ from the lower working chamber into the upper working chamber can only flow through the compression stage valve in the shock absorber piston with low flow resistance. This prevents a radical drop in pressure in the upper working chamber. In contrast, in the rebound stage, i.e. when the shock absorber is moving out, the damping work is essentially done by the volume flow $\dot{Q}_1$ from the upper working chamber into the lower working chamber at the corresponding piston valve. The bottom valve only balances out the extending

piston rod volume in that shock absorber oil flows virtually without resistance from the compensating chamber into the lower working chamber (volume flow $\dot{Q}_2$).

The compensating chamber means that twin-tube shock absorbers have poorer heat dissipation in comparison with single-tube shock absorbers. Furthermore, the installation position of twin-tube shock absorbers is restricted, as it must be ensured at all times that there is compensating fluid at the bottom valves. Advantages compared to the single-tube shock absorber are the lower shock absorber length, the softer responsiveness, as well as the lower requirements with regard to seals. In the area of passenger cars, the twin-tube shock absorber has prevailed as the standard shock absorber – also due to its lower costs.

Adjustable shock absorbers

The conflict of objectives with regard to the coordination of body shock absorbers between driving comfort and driving safety can be mitigated using adaptive or semi-active shock absorbers. In comparison with passive shock absorbers with fixed shock absorber characteristics (i.e. defined force-speed characteristics, cf. section "Damping characteristics"), adaptive shock absorbers provide the possibility of discrete to infinitely variable adjustment of the damping characteristics (Figure 2).

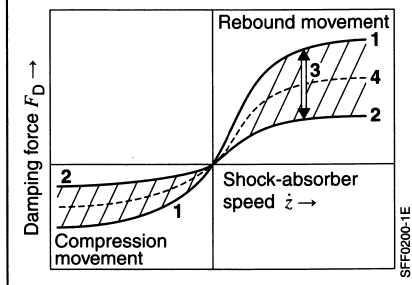
Alongside manual adjustment of the shock absorbers (for example soft damping in the comfort mode or hard damping in the sport mode), adjustable shock absorbers can also be activated automatically depending on the driving state in each case (see section "Damping control").

Adaptive hydraulic absorbers

In the case of adaptive or semi-active shock absorbers of conventional design, the adjustability of the damping characteristics is implemented by means of adjustable shock absorber valves, bypass boreholes that can be activated (located on the outside or inside), or with the help of double pistons [1]. As a rule, activation is electronic. Major system features are the adjustment times that can be achieved, the spread of the adjustment range, as well as

Figure 2: Adjustable shock absorber characteristic curves

- 1 Upper limit of the adjustment range (maximum damping hardness, for example sport mode),
- 2 Lower limit of the adjustment range (minimum damping hardness, for example comfort mode),
- 3 Adjustment range of the shock absorber,
- 4 Characteristic curve of a passive shock absorber.



the number of definable shock absorber characteristics. Whereas systems of the first generation only permitted adjustment between a few characteristic curves, today's adaptive shock absorbers can usually be set to a large number of characteristics [2]. Newer systems usually even have infinitely variable adjustment between a minimum and maximum shock-absorber force characteristic (Figure 2).

Rheological shock-absorber systems

The adjustability of the damping characteristics in the case of rheological shock-absorber systems is based on the change in the flow properties of the working medium that is used. Magneto-rheological or electro-rheological fluids that change their viscosity under the influence of a magnetic or electrical field are used here instead of the usual mineral oils. The viscosity of the working medium has a direct influence on the flow resistance through the shock absorber valves. If, for example, creating a magnetic field increases the viscosity of a magneto-rheological working medium, the flow resistance through the shock absorber valves increases. Rheological shock-absorber systems not only provide the possibility for infinitely variable adjustment of the damping char-

acteristics but also the implementation of very short adjustment times [2].

Damping characteristics

Damping force is a function of the speed of the inward or outward movement of the shock absorber, whereby the direction of force is opposed to the direction of speed at all times. It is generally applicable that the damping force F_D and the speed $\dot{z}$ are linked via the damping constant k_D and the damping exponent n . The following applies:

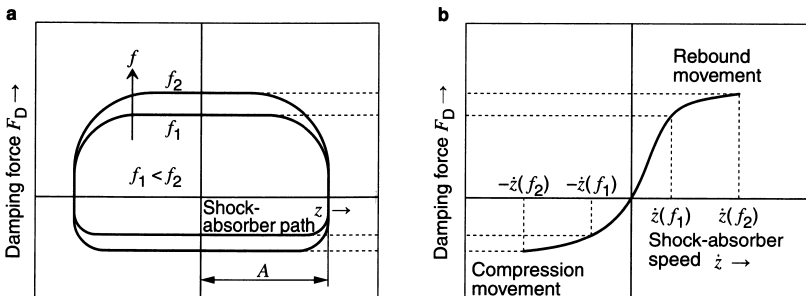
$$F_D = -\text{sign}(\dot{z}) \cdot k_D \cdot |\dot{z}|^n$$

The damping constant and damping exponent are essentially dependent on the design of the shock absorber (valve characteristics, geometry). With the corresponding configuration of the individual parameters, progressive to degressive damping characteristic curves can be created. Modern body shock absorbers have mainly degressive characteristics. This achieves a high damping effect at low excitation speeds as well as a limitation of the maximum shock-absorber forces.

Damping characteristic curves are usually determined with the help of mechanical or servo-hydraulic testing units.

Figure 3: Damping characteristics

- a) Work diagram (force-path diagram),
- b) Damping characteristic curve (force-speed diagram).
- f Variable excitation frequency,
- f_1 Excitation frequency 1,
- f_2 Excitation frequency 2,
- $\dot{z}(f_1)$ Maximum shock-absorber speed with f_1 ,
- $\dot{z}(f_2)$ Maximum shock-absorber speed with f_2 ,
- A Constant vibration amplitude.



A sinusoidal path excitation of constant amplitude and variable frequency or constant frequency and variable amplitude results in various maximum speeds of the inward or outward movement. The recorded path and force signals can be applied in a force-path diagram (work diagram) (Figure 3a). The force-speed characteristic curve of the shock absorber (damping characteristic curve) can be derived from the work diagram by transferring the maximum force and speed values (Figure 3b).

It is mainly for reasons related to comfort that the configurations of the rebound and compression stage differ. The shock-absorber forces created in the rebound stage are usually more than double the correspondingly created forces on compression (i.e. in the compression stage) of the shock absorber (Figure 3). This limits the impact forces on the vehicle body

during the compression phase (comfort) and simultaneously ensures the system is strongly damped (system relaxation) in the rebound phase.

Damping control

In conjunction with electronically adjustable shock absorbers, damping control systems are being used to an increasing degree nowadays. The major component parts of such damping control systems are the adaptive shock absorbers and sensors (for example acceleration sensors on the wheel and body mass) and intelligent algorithms and control strategies. With the help of the sensors and algorithms, the current driving state is continuously determined and evaluated. In accordance with the stored control strategies, this enables the control system to adapt the shock-absorber characteristics to each driving state by activating the shock absorbers, thus, for example, influencing and optimizing driving comfort or driving safety.

Control strategies

Threshold-value strategy

Threshold-value controllers compare one or a number of relevant driving state variables (for example body acceleration, steering angle) with the corresponding threshold values and initiate defined measures if the threshold values are exceeded or not reached. The shock-absorber forces are usually influenced axle by axle simultaneously in the rebound and compression directions. The main focus of threshold-value controllers is on increasing comfort while simultaneously maintaining driving safety.

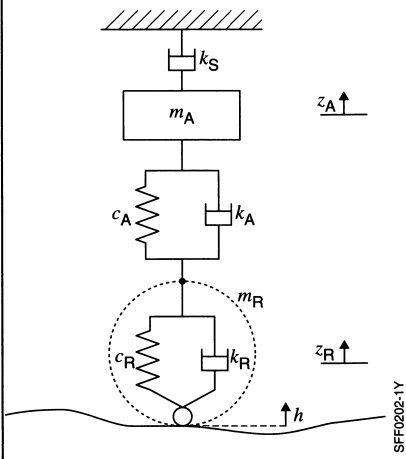
Alongside purely influencing the vertical vibration characteristics, induced vehicle-body movements can also be optimized. For example, steering angle monitoring can lead to a reduction in the dynamic rolling or hardening the damping depending on the brake pressure can reduce pitching motion caused by braking.

Skyhook

The skyhook control strategy aims to keep the vehicle body calm independently of current driving state and road conditions. This is intended above all to increase driv-

Figure 4: Theoretical principles of the skyhook approach

- k_S Damping constant of the sky shock absorber,
- m_A Vehicle-body mass,
- z_A Vertical vehicle-body motion,
- c_A Spring rigidity of the vehicle-body springs,
- k_A Damping constant of the vehicle-body shock absorber,
- m_R Wheel mass (unsprung mass),
- z_R Vertical wheel movement,
- c_R Vertical tire spring rigidity,
- k_R Damping constant of the tire,
- h Vertical road excitation.



ing comfort. In contrast to the threshold-value strategy, the skyhook control strategy regulates the damping characteristics at each individual wheel. The basic principle is to decouple the movement of the vehicle body from the road excitation. To achieve this, it is imagined that the vehicle body is connected by means of a shock absorber to the sky (Figure 4). The shock-absorber force F_{DS} of the skyhook shock absorber results from the link of the body speed $\dot{z}_A$ and the damping constant k_S of the imaginary sky shock absorber:

$$F_{DS} = k_S \dot{z}_A .$$

For the conventional vibration system, on the other hand, the shock-absorber force F_D would result from the link of the damping constant k_A of the vehicle-body shock absorber and the difference between the vertical body speed $\dot{z}_A$ and the vertical wheel speed $\dot{z}_R$:

$$F_D = k_A (\dot{z}_A - \dot{z}_R) .$$

In order to brace the vehicle body against the sky, in the real implementation the additional portion of force F_{DS} of the sky shock absorber must be applied by the body shock absorber. The proportional

damping factor k_{AS} this requires is calculated to:

$$k_{AS} = \frac{k_S \dot{z}_A}{\dot{z}_A - \dot{z}_R} .$$

As an adaptive (semi-active) shock absorber is only able to extract energy from the system in the form of heat, but is unable to supply heat to the system, a case distinction is required [1], [2]. The following applies:

$$F_{Dtot} = \left(k_S \frac{\dot{z}_A}{\dot{z}_A - \dot{z}_R} + k_A \right) \cdot (\dot{z}_A - \dot{z}_R)$$

where $\dot{z}_A (\dot{z}_A - \dot{z}_R) \geq 0$, and

$$F_{Dtot} = k_A (\dot{z}_A - \dot{z}_R)$$

where $\dot{z}_A (\dot{z}_A - \dot{z}_R) < 0$.

Depending on the amount and direction of the vehicle-body speed and the shock absorber movement (rebound or compression movement), the control strategy for the skyhook approach shown in Figure 5 is pursued for comfortable damping of the vehicle body. However, specific damping of rolling, pitching, and wheel vibrations is not taken into account by this approach. These movements are also highly relevant with regard to driving comfort and

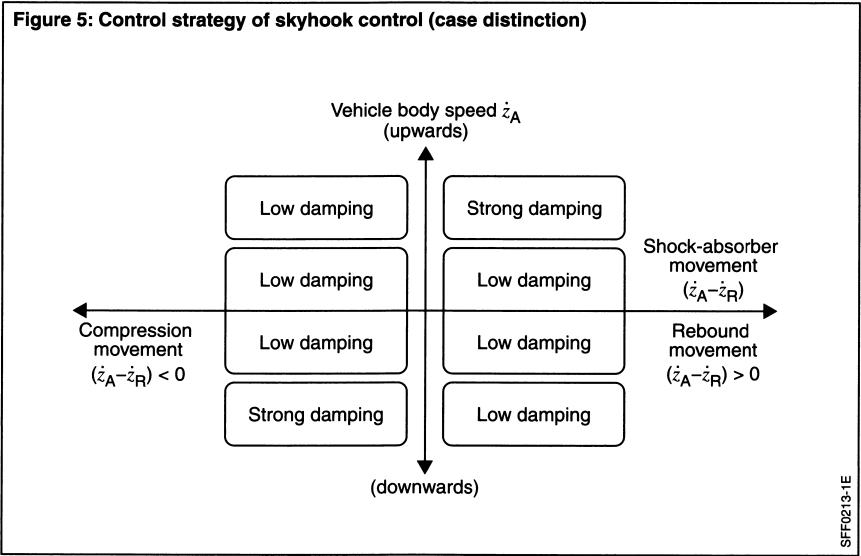
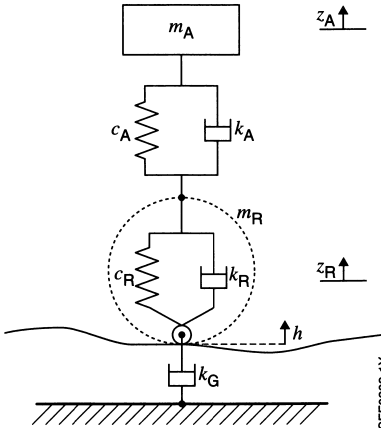


Figure 6: Theoretical principles of the groundhook approach

m_A Vehicle-body mass,
 z_A Vertical vehicle-body motion,
 c_A Spring rigidity of the vehicle-body springs,
 k_A Damping constant of the vehicle-body shock absorber,
 m_R Wheel mass (unsprung mass),
 z_R Vertical wheel movement,
 c_R Vertical tire spring rigidity,
 k_R Damping constant of the tires,
 h Vertical road excitation,
 k_G Damping constant of the groundhook shock absorber.



driving safety, which means that the skyhook controller is usually overridden by other controllers.

Groundhook

A groundhook controller aims to improve driving safety by reducing the wheel load fluctuations. In much the same way as the considerations for the skyhook strategy, it is imagined that the wheel is connected by a shock absorber to the roadway (Figure 6) and a proportional damping factor k_{AG} is derived. The following applies in the same way as the derivations for the skyhook controller:

$$k_{AG} = k_G \frac{\dot{z}_R - \dot{h}}{\dot{z}_R - \dot{z}_A}$$

where:

$\dot{z}_A$ Vertical body speed,

$\dot{z}_R$ Wheel speed

$\dot{h}$ Vertical excitation speed,

k_G Damping constant.

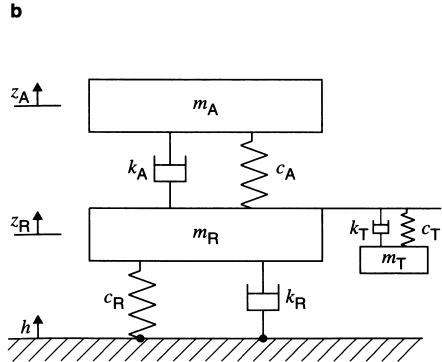
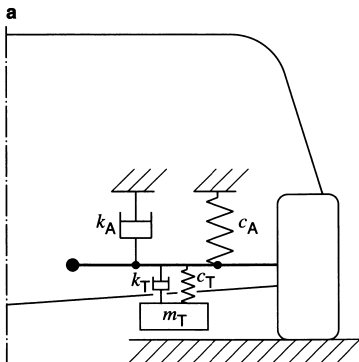
Also for the groundhook controller, a case distinction is made depending on the directions of movement of the wheel and vehicle body. This case distinction takes place on the basis of the term:

$$(\dot{z}_R - \dot{h})(\dot{z}_R - \dot{z}_A).$$

Figure 7: Vibration absorber

a) Vibration absorber in the chassis (diagram), b) Substitute system.

k_A Damping constant of the vehicle-body shock absorber,
 c_A Spring rigidity of the vehicle-body springs,
 k_T Damping constant of the vibration absorber, c_T Spring rigidity of the absorber springs,
 m_T Absorber mass, m_A Vehicle-body mass, z_A Vertical vehicle-body motion, m_R Wheel mass,
 z_R Vertical wheel movement, c_R Vertical tire spring rigidity,
 k_R Damping constant of the tire, h Vertical road excitation.



Vibration absorbers

To specifically influence the vibration properties of the vibration system consisting of the wheel and vehicle-body mass, vibration absorbers (see chapter "Vibrations and oscillations") are deployed in some cases in the area of the chassis.

Depending on the configuration and arrangement of the vibration absorber, the comfort, acoustics, or driving safety can be influenced. A distinction is made between passive and active vibration absorbers. A passive vibration absorber is a mass attached to the chassis by sprung and damped mountings (Figure 7). The absorbing effect is created by the corresponding mass forces and in the case of passive absorbers is restricted to a certain frequency range. The effective range can be enlarged using an active absorber with an actuator that can be activated.

On excitation of the vibration system, the vibrations and oscillations of the main system are taken over by the appropriately coordinated vibration absorber, i.e. the main system only vibrates very slightly whereas the absorber absorbs a large portion of the energy. Figure 8 shows an example of the progression of the vibration amplitude of a wheel movement with

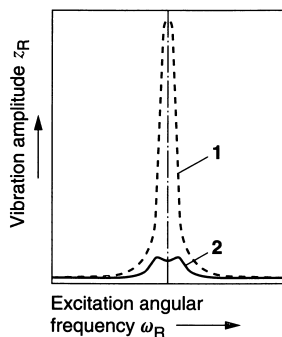
and without vibration absorbers. With the deployment of a vibration absorber geared to the frequency range of the natural wheel frequency, a significant drop in the vibration amplitude can be observed in the corresponding frequency range.

References

- [1] B. Heissing, M. Ersoy (Editors): *Fahrwerkhandbuch*. 1st Edition, Vieweg Verlag, 2007.
- [2] L. Eckstein: *Aktive Fahrzeugsicherheit*. ika/fka 2010.

Figure 8: Vibration amplitude of a wheel movement as a function of the excitation angular frequency with and without vibration absorbers

- 1 Progression without vibration absorber,
 - 2 Progression with vibration absorber.
- ω_R Natural angular frequency of wheel.



Wheel suspensions

Basic principles

Vehicle wheels and the vehicle body are connected via wheel suspensions. On the one hand, a wheel suspension has the function of guiding the relevant wheel in relation to the vehicle body in such a way that a movement that is essentially directed vertically relative to the vehicle body remains possible and, on the other hand, that the tire forces exerted in the wheel contact point in the horizontal plane and the torques generated by these forces can be transferred to the body. On the front axle and also on the rear axle of vehicles with rear-axle steering, additional steerability of the wheels is to be ensured.

Alongside the tires, the suspension and shock absorber system, the vehicle body mass, and the individual wheel masses, the wheel suspensions have a major influence on the driving characteristics of a vehicle, as they influence the parameters of the vehicle axle concerned that are relevant to driving dynamics. These are, for example:

- the track width,
- the toe-in or toe-out angle,
- the camber angle,
- the caster angle,
- the caster offset in wheel center,
- the spread angle,
- the spread offset,
- the kingpin offset,
- the disturbing force lever arm,
- the position of the rolling pole of the axle and thus the orientation of the roll axis,
- the location of the pitching pole,
- the braking and anti-squat control,
- the longitudinal and transversal springs.

For the definitions of the individual wheel suspension or vehicle parameters, refer to the chapter “Basic terms of automotive engineering:”.

Kinematics and elastokinematics

During operation of the vehicle, the geometry and kinematics of a wheel suspension lead to changes in the characteristic parameters of the wheel suspension (for example, caster angle, location of the rolling pole) and wheel-position parameters (for example, camber angle and toe angle) of the corresponding wheel. This is the case, for example, as a result of a movement of the wheel within the framework of the vertical degree of freedom permitted by the wheel suspension (i.e. compression or rebound movement of the wheel) or with steerable wheel suspensions as a result of a steering movement. Figure 1 shows examples of the camber- and toe-angle changes of a wheel suspension due to kinematics during compression by the compression path Δz in comparison with the design position. As a rule, the kinematic wheel-position changes are shown with the help of diagrams of the compression and rebound path of the wheel. The so-called “wheel paths” for the camber- and toe-angle changes of the wheel suspension shown in Figures 1a and 1b are shown in Figure 1c.

Due to the fact that the kinematic changes to the wheel-suspension parameters and wheel-position parameters have a great influence on the driving characteristics of a vehicle, correct configuration and coordination of the steering and wheel lift kinematics are of great significance.

Alongside the changes to the wheel-position parameters due to kinematics during compression or rebound movements, the forces and torques affecting the wheel suspension (for example drive and braking forces, lateral and vertical forces in the wheel contact point) in conjunction with the elasticity of the suspension lead to further wheel-position changes. The elasticity of a wheel suspension results from the deformability of the individual wheel suspension components (for example

links) and the bearings used when forces and torques are applied. For reasons related to driving comfort and acoustics, the mounts used in modern wheel suspensions are usually elastic mounts (for

example rubber mounts). Figure 2 shows an example of a wheel suspension with two rubber mounts fitted on the vehicle body side; their elasticity when longitudinal force occurs at the wheel contact point leads to an elastokinematic change in the toe angle of the wheel.

Alongside the kinematic wheel-position changes, the elastokinematic effects also influence the driving characteristics of a vehicle. When coordinating the kinematics and elastokinematics of a wheel suspension, this is why it is usually the aim that the kinematic and elastokinematic effects supplement one another when influenced by forces and springs.

For example, elastokinematic steering is used on a number of modern rear-axle suspensions to reduce load change reactions (for example by increasing the toe-in when braking force affects the rear wheel on the outside of the bend) [1]. It is possible to influence the elastokinematic properties of a wheel suspension by, for example, coordinating the individual mount elasticities or adjusting individual mounting points.

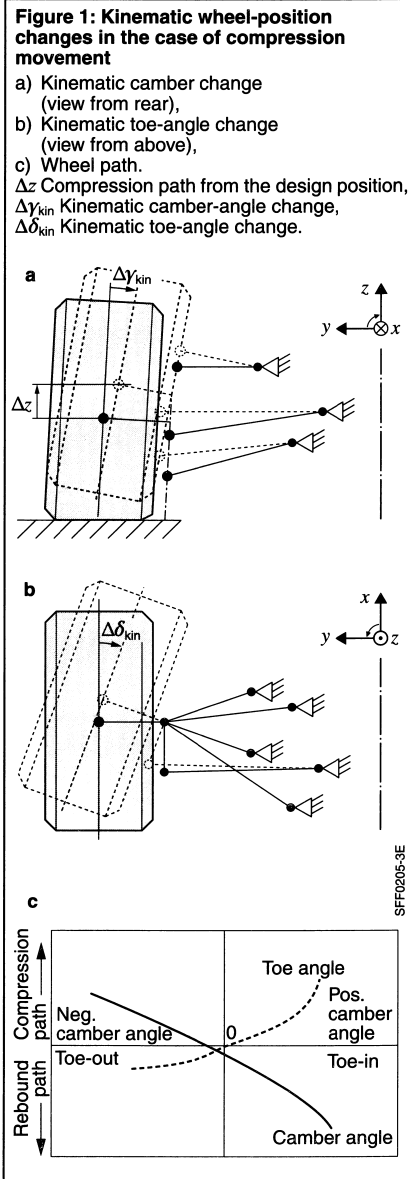
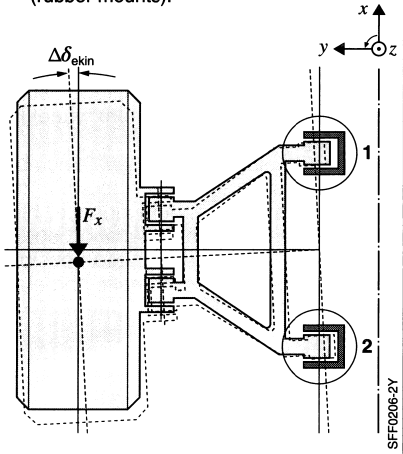


Figure 2: Elastokinematic toe-angle change due to effect of longitudinal force

F_x Longitudinal force (braking force),
 $\Delta \delta_{ekin}$ Elastokinematic toe-angle change,
1 Elastic shift of pivot point 1 (rubber mounts),
2 Elastic shift of pivot point 2 (rubber mounts).



Basic categories of wheel suspensions

There are a large number of different wheel suspensions. They are classified primarily according to the type of suspension concept. An initial distinction is made between rigid axles (dependent wheel control), semi-rigid axles, and independent wheel suspensions (independent wheel control).

Rigid axles

In the case of a rigid axle, the wheels of an axle are firmly interconnected by a rigid axle body, which leads to mutual influences on the wheels. Rigid axles are used as both driven and non-driven rear axles on heavy vehicles (for example off-road vehicles, light utility vans, trucks). Occasionally, however, their sturdy construction and high ground clearance mean that steerable variants are also used as front axles (for example on off-road vehicles or off-road trucks).

Guidance of a rigid axle in relation to the vehicle body can be implemented in different ways. On vehicles with leaf springs, guidance is usually via the spring leaves (Figure 3a). There are also a large number of rigid axle concepts guided by links or coupling shafts (Figures 3b, 3c, and 3d). Where links and coupling shafts are used, statically undefined mounts are selected to make linking at the vehicle body easier and to reduce the required space [3]. Refer to [2] for detailed explanations of the individual axle variants.

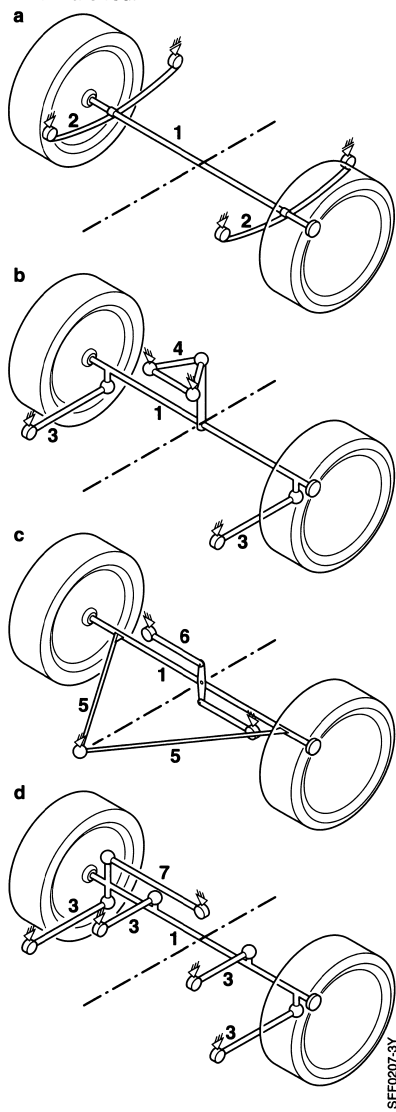
The major advantages of rigid axles are the simple and sturdy design, low costs, a high roll center, high maximum wheel lift, and high ground clearance.

However, rigid axles also have a number of disadvantages that are inherent in the design: the mutual wheel influence, high unsprung mass, high installation space requirement, as well as limited possibilities to coordinate the kinematic and elastokinematic factors.

Figure 3: Design examples for rigid axles

- a) Longitudinal leaf-spring suspension,
b) Trailing and coupling-shaft links,
c) Coupling shaft with watt linkage,
d) Trailing link with panhard rod.

- 1 Rigid axle, 2 Leaf spring,
3 Trailing link, 4 Coupling-shaft link,
5 Coupling shaft, 6 Watt linkage,
7 Panhard rod.



Semi-rigid axles

Semi-rigid axles also involve a mechanical coupling of the wheels. In contrast to rigid axles, however, this coupling is not rigid. The elasticity of the coupling profile that is used enables relative movements between the wheels. The coupling profile forms a cross-connection between two trailing links to which it is firmly connected. Longitudinal forces are absorbed via the trailing links. The bracing of lateral forces is supported by the stiffening effect of the coupling profile. In order to guarantee a relative movement between the two wheels of the axle, the coupling profile has to be designed as weak. Depending on the arrangement of the coupling profile, distinctions are made between torsional-link axles, twist-beam axles and semi-independent axles (Figure 4).

Due to their simple and low-cost design, semi-rigid axles are widely used as rear axles on vehicles with front-wheel drive.

The advantages of this axle concept include the low installation space requirement, the low unsprung masses, easy assembly and removal, the stabilizing effect of the coupling profile, the low track width and toe-angle changes, as well as the good anti-dive properties.

These advantages are offset by a number of disadvantages that are inherent in the principle: the mutual wheel influence, the low suitability for driven axles, the high tension peaks at the transition points between the trailing link and coupling profile, the increase in the tendency to oversteer in the event of influencing lateral forces (lateral force oversteer) due to link deformations, as well as limited kinematic and elastokinematic optimization potential.

Torsional-link axles

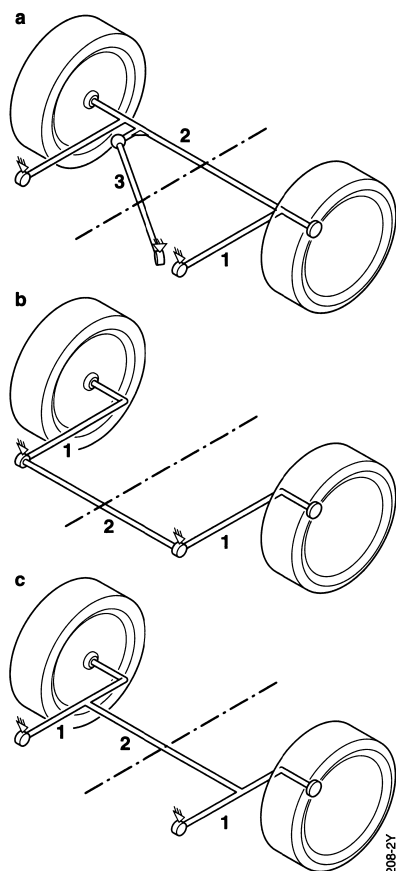
In the case of torsional-link axles (Figure 4a), the two wheel carriers are connected by means of a coupling profile arranged close to the wheel center. As a rule, the lateral guidance of the axle is supported by an additional guide element (for example a panhard rod) [2]. There are great similarities to a rigid axle with regard to both the structure and properties.

Twist-beam axles

In contrast, twist-beam axles (Figure 4b) have kinematic properties similar to those of trailing-link wheel suspensions. The coupling profile is arranged at the height of the pivot points of the trailing link. The deployment and arrangement of the cou-

Figure 4: Design examples for semi-rigid axles

- a) Torsional-link axle with panhard rod,
 - b) Twist-beam axle,
 - c) Semi-independent axle.
- 1 Trailing link,
2 Coupling profile,
3 Panhard rod.



pling profile greatly simplifies the mounting of the trailing links compared to a trailing-link independent wheel suspension.

Semi-independent axles

In comparison with twist beam axles, the coupling profile on a semi-independent axle (Figure 4c) is not at the height of the link pivot points rather is offset towards the rear. This improves above all lateral force bracing in comparison with the twist-beam axle.

Independent wheel suspensions

Alongside semi-rigid axles – as rear axles on vehicles with front-wheel drive – most modern vehicles nowadays have independent wheel suspensions where each wheel is individually connected to the vehicle body according to the desired degrees of freedom of movement. A wheel is connected here with the help of a wheel carrier and a corresponding number of links.

The design (for example two-point links or A-arm links) and arrangement of the links (trailing links, transverse links or diagonal links) and the connecting mounts determine the kinematic and elastokinematic properties of the wheel suspension. The design of the individual links determines the number required to reduce the freedom of movement of a wheel to the desired number of degrees of freedom.

The number of links is frequently used as the classification of suspension types (for example, five-link independent wheel suspension). The resulting type of spatial movement (kinematics) of the wheel during compression and rebound movement is also frequently used for the classification of independent wheel suspensions [1], [3]. Depending on the type of movement of the wheel carrier, distinctions are made here between level, spherical, and spatial independent wheel suspensions [1], [3].

The proportion of independent wheel suspensions in modern vehicles is rising steadily. Compared to rigid and semi-rigid axles, independent wheel suspensions offer a number of advantages. For example, there is no mutual wheel influence, the kinematic and elastokinematic optimi-

zation potential is high, and the space requirement and unsprung mass are low in some cases.

However, independent wheel suspensions also have some disadvantages. In some cases, they lead to a complex design, the costs are high, the maximum wheel lift is low, and the configuration and coordination process is more complex in some cases.

There are large numbers of different designs of independent wheel suspensions. The basic principles of selected designs are to be explained briefly in the following section and their structures shown in diagrams. For detailed explanations of the individual designs and of other independent wheel suspensions and specific design examples, refer to [2].

Trailing-link independent wheel suspension

On a trailing-link independent wheel suspension, a wheel is connected to the vehicle body by means of a single link arranged in longitudinal direction (Figure 5a). The trailing link transfers both the longitudinal and lateral forces, which means that high mount forces occur and the mounts have to be designed accordingly.

The axis of rotation of the links runs parallel to the vehicle transverse axis. The advantages of this suspension form are usually low installation space requirement as well as low costs. The disadvantages are the limited kinematic optimization possibilities, the instantaneous center located at road height that causes high roll torque on cornering, as well as the high stresses placed on the links and their mounts.

Diagonal-link independent wheel suspension

In the same way as on the trailing-link wheel suspension, on the diagonal-link wheel suspension the wheel is also connected to the vehicle body by means of a single link. However, to achieve better bracing of the longitudinal and above all lateral forces, the link is arranged diagonally (Figure 5b) and there is more space between the mounting points. To achieve more favorable kinematic properties, on modern suspensions the axis of rotation of the link is arranged diagonally both in

Figure 5: Trailing- and diagonal-link independent wheel suspension

- a) Trailing-link independent wheel suspension,
b) Diagonal-link independent wheel suspension.
1 Trailing link,
2 Diagonal link.

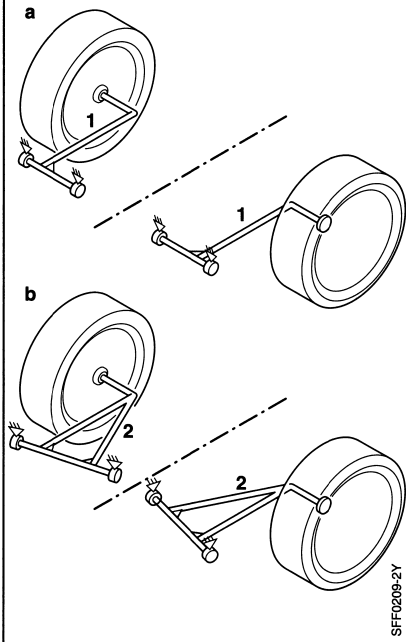
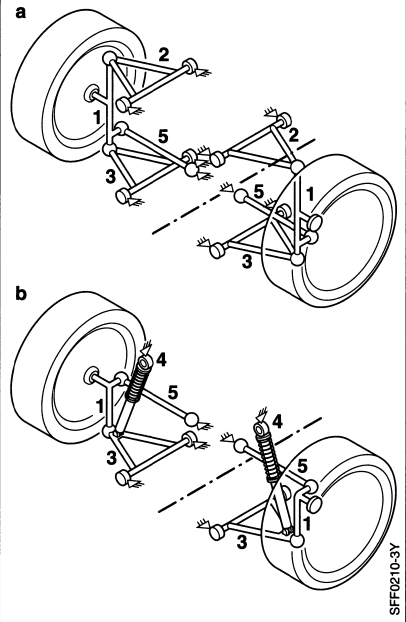


Figure 6: Double-wishbone and spring-strut independent wheel suspension

- a) Double-wishbone independent wheel suspension,
b) Spring-strut independent wheel suspension.
1 Wheel carrier, 2 Upper A-arm link,
3 Lower A-arm link,
4 Spring shock absorber strut,
5 Tie rod (steering).



the projection to the vehicle lateral plane (roof angle) and in the projection to the road (V-shaped angle) [1].

Double-wishbone independent wheel suspension

A double-wishbone independent suspension is when a wheel is connected to the vehicle body via two A-arm links. One link is arranged below the wheel center and the other is arranged above it (Figure 6a), which enables the suspension to brace all of the forces and torques that occur at the wheel. As a rule, the high articulation forces mean that the transverse links are not directly connected to the vehicle body structure but rather are secured to a "chassis subframe" that interconnects

both wheel suspensions and thus relieves the load of inner forces on the vehicle body.

With adaptation of the mounts and design of the links, double-wishbone independent suspensions provide very high kinematic optimization potential [2]. Depending on the location of the axes of rotation of the links, level, spherical or spatial wheel suspension kinematics are achieved [3]. The disadvantages of double-wishbone independent suspensions are the higher costs as well as the greater installation space requirement.

Spring-strut independent wheel suspension

The kinematics of a spring strut wheel suspension correspond to those of a double-wishbone independent suspension in which the upper transverse link is replaced by a sliding guide (Figure 6b). This sliding guide corresponds to the spring strut on other suspensions (in the case of a combined spring-shock absorber unit) or the shock absorber strut (in the case of separate spring and shock absorber arrangement) where the housing is rigidly connected to the wheel carrier. With this design, the shock absorber rod also assumes wheel guidance tasks.

The lower link plane of a spring-strut suspension is usually formed by two two-point links (radius links) or an A-arm link. In the case of a suspension using the McPherson principle, originally the lower A-arm link was formed from a transverse link and a stabilizer. Nowadays, however, other spring-strut suspensions are also referred to as McPherson axles.

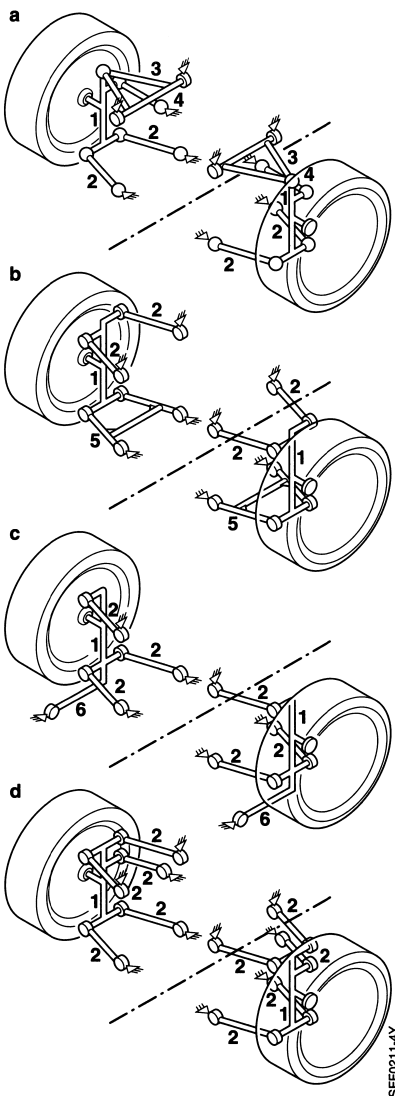
Advantages of the spring-strut wheel suspension are above all the low construction overhead and low amount of space required at the height of the wheel axes which can be used in particular in the case of passenger cars with front-wheel drive with laterally installed engine-transmission units. Other advantages are the cost-saving and weight-saving design, simple installation and assembly, as well as the high degree of integration. On specifying the kinematics, however, spring-strut suspensions offer slightly less leeway for configuration in comparison with double-wishbone independent suspensions.

Multilink independent wheel suspension

Wheel suspensions with four or five single links are generally referred to as multilink axles. Multilink axles result, for example, from breaking down an A-arm link into two individual two-point links in the case of a double wishbone axle (Figure 7a). Breaking down three-point links and the use of independent two-point links usually result in greater leeway for configuration of the kinematic and elastokinematic properties of the axle. On the one hand, this increases the optimization of the

Fig. 7: Design examples for multilink independent wheel suspensions

- a) Multilink independent wheel suspension,
 - b) Trapezoidal-link independent wheel suspension,
 - c) Control-blade independent wheel suspension,
 - d) Five-link independent wheel suspension.
- 1 Wheel carrier, 2 Two-point link, 3 A-arm link, 4 Tie rod (steering), 5 Trapezoidal link, 6 Control blade.



axle with regard to comfort and driving safety requirements. On the other hand, the somewhat complex design increases the overhead involved in the configuration and coordination processes for the wheel suspension.

Trapezoidal-link independent wheel suspension

Trapezoidal-link wheel suspensions (Figure 7b) are a special form of the multilink independent wheel suspension that is mainly used as rear axles. The lower plane is formed by a trapezoidal link that has two connection points on the wheel side and an axis of rotation on the vehicle body side. This means the lower link sets a total of three degrees of freedom of the wheel. Another two degrees of freedom are eliminated by two two-point links arranged accordingly so that only the desired compression degree of freedom of the wheel remains.

Control-blade independent wheel suspension

Another design of multilink independent wheel suspensions is the control blade axle on which the wheel is guided by one trailing link and three transverse links (Figure 7c). The trailing link (control-blade link) is connected to the vehicle body in such a way that it can be rotated and is

firmly connected to the wheel carrier. As a rule, it has elastic properties in order to enable kinematic track and camber changes. The lateral forces are braced via three two-point transverse links which are usually arranged in two planes (one above and one below the center of the wheel). The arrangement and orientation of the transverse links determines the kinematic properties of the wheel suspension.

Five-link independent wheel suspension

A wheel suspension with completely detached links requires five individual two-point links to reduce the movement of a wheel to the desired vertical degree of freedom (Figure 7d). Five-link rear axles are generally referred to as multilink suspensions, whereby on the front axle this is referred to as a four-link axle plus tie rod [2].

References

- [1] L. Eckstein: Vertikal- und Querdynamik von Kraftfahrzeugen. ika/fka 2010.
- [2] B. Heissing, M. Ersoy (Editors): Fahrwerkhandbuch. Vieweg+Teubner Verlag, 2008.
- [3] M. Matschinsky: Radführungen der Straßenfahrzeuge. Springer-Verlag, 2007.

Wheels

Function and requirements

All of the vehicle-specific or axle-specific tasks are performed via the wheel, e.g. transfer of dynamic forces between the vehicle and road surface. These include taking up the vehicle load and the impact forces of the road surface, transferring the rotary motion of the axles to the tires, and taking up and transferring braking and acceleration forces as well as lateral forces when cornering. Wheel size is mainly determined by the space required by the braking system, the axle components, and the size of the tires used.

Wheels have primarily a technical function. But the increasingly booming light-alloy wheel market calls for visually attractive designs.

Structure

The wheel is a load-bearing, rotating part between the tire and the axle. It usually consists of two main components – the rim and the wheel disk. These two components can be made from a single part, and can also be permanently or non-permanently attached to each other. A permanent connection of a rim with a wheel disk is called a disk wheel.

Figure 1 shows the basic structure of a steel wheel. Here the rim mounts the tire and the wheel disk connects the wheel to the axle.

In daily usage the terms rim and wheel are often interchanged. The term rim is often used when actually the complete wheel is meant. The “wheel” in general usage often also refers to the tire. However, as a technical term in automotive engineering “wheel” generally means the wheel without the tire.

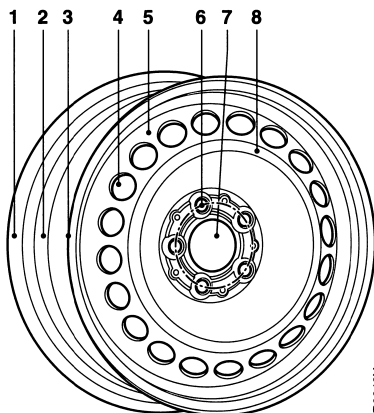
Wheel disk

The wheel disk (wheel nave) is the part that connects the rim to the axle hub. In the case of a steel wheel, the wheel disk consists of a formed steel-sheet blank. This has holes for ventilating the brake system and is usually curved (dish, see Figure 1). The center of the wheel disk contains the center hole and the wheel-bolt or stud holes. The wheel is secured to the axle through these holes. The center hole is provided with a locating bore by means of which the wheel is radially centered on the axle. This bore determines together with the rim bead seat (for the bead ledge) the wheel's true-running quality (via the radial run-out). The attachment face is, together with the rim flanges, responsible for lateral wheel run-out.

In the case of an aluminum wheel, the wheel disk can, due to its design, have different shapes such that the term wheel disk is no longer applicable.

Figure 1: Structure of a disk wheel

- 1 Rim inner flange,
- 2 Rim base,
- 3 Rim outer flange,
- 4 Ventilation hole,
- 5 Wheel disk,
- 6 Pitch-circle diameter,
- 7 Center hole,
- 8 Dish.



Rim

The term rim strictly speaking describes only the radially outermost part of the wheel which holds the tire. The rim is therefore the fundamental connecting element between the wheel disk and the tire. In a tubeless tire it provides the air seal and is geometrically matched to the tire. The most commonly used shape of rim is divided into four zones (Figure 2):

- rim flange (inner and outer),
- hump (inner and outer),
- rim bead seat (inner and outer),
- rim base and drop center.

There are also different shapes for passenger cars (Figure 3).

Rim flange

The rim is limited on the inner and outer sides by a rim flange (rim inner flange and rim outer flange). It acts as the side stop for the tire bead (see Tires) and absorbs the forces resulting from the tire pressure and the axial tire load. The rim flange is specified in the guidelines of the ETRTO (European Tyre and Rim Technical Organisation) by, for example, J, K, JK or B. In this way, the geometry of the rim flange and the ratio to the drop center are dimensionally described. This is based on how the wheel is used.

The most common rim-flange shape for passenger cars is the J flange shape. The lower B flange can be found on smaller vehicles and in inflatable spare tire systems. K and JK flanges are rarely used any more; they used to be the domain of

heavy, comfortable vehicles of the upper price segment.

Rim bead seat

The rim bead seat describes the contact zone of the tire with the rim. It centers the tire radially. In this zone the tire is given the correct position for radial and lateral true running. All the dynamic driving forces are transferred here. In the tubeless tires mainly used today in passenger cars the wheel/tire system is sealed at the rim bead seat.

Figure 3: Rim systems for passenger cars

- a) Passenger-car drop-center rim (standard rim),
 - b) EH2+ rim (extended hump),
 - c) PAX rim (Pneu Accrochage, X stands for Michelin radial-tire technology),
 - d) CTS rim (Conti Tire System), steel-wheel version.
- 1 Rim flange, 2 Tapered bead seat, 3 Hump, 4 Drop center.
M Flange-to-flange width,
D Nominal rim diameter,
D_H Hump diameter.

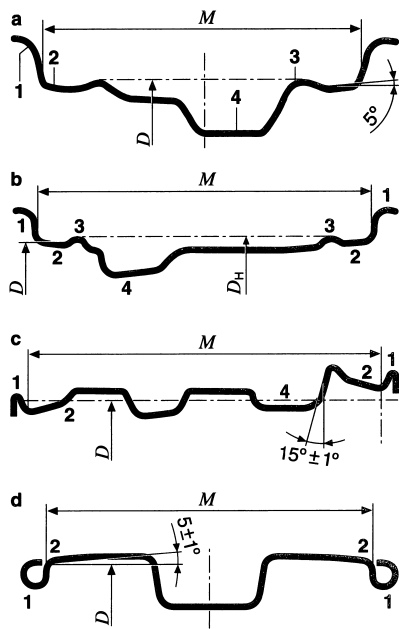
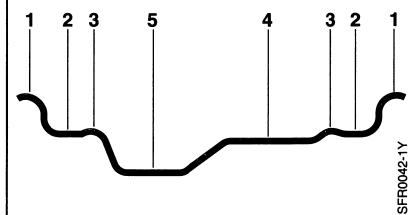


Figure 2: Rim zones

- 1 Rim flange,
- 2 Rim bead seat,
- 3 Hump,
- 4 Rim base,
- 5 Drop center.



Rim base

The rim base connects the inner and outer rim bead seats. The drop-center rim is primarily used in passenger cars. Drop-center rims have a clearly defined shape with a drop-center rim base (drop center). When the tire is mounted on the wheel, the tire is initially positioned with one side of the tire bead in the drop center so that it can be pulled over the rim flange on the opposite side. The drop center is the required shaping of the rim base to mount the tire base (tire bead, tire inner ring) when fitting and removing the tire.

Hump

The hump is the all-round raised bead in the area of the rim bead seat (Figure 4). It is prescribed in many countries for tubeless tires. In the event of low tire pressure the hump is intended to prevent the tire from coming off the rim bead seat. The following hump shapes are customary:

- H: round hump on one side on the outer bead seat,
- H2: round hump on both sides,
- FH: flat hump on the outer bead seat,
- CH: combination hump, flat hump on the outer bead seat, round hump on the inner bead seat,
- EH2: extended hump on both sides.

Primarily H2 rims are used in passenger cars. H rims (formerly also called H1) are also used in commercial vehicles and older vehicles as well. The major differences are between the standard hump shape (H) and the flat hump (FH). Newly included in the standard is the extended hump (EH2) with slightly larger hump diameter, which is used in some cases, particularly with "run flat" tire systems.

Rim and wheel dimensions

Terms

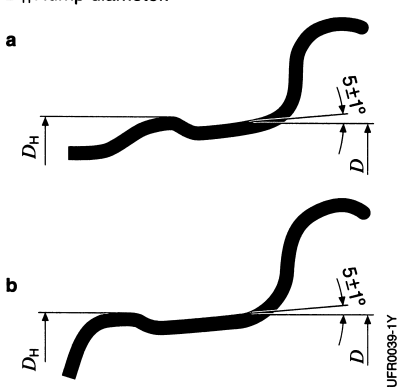
The most important terms for the function and design of a wheel are (Figure 5):

- rim diameter (nominal diameter, dimension from rim bead seat to rim bead seat),
- rim circumference (measured value, determined with bead tape around the rim bead seat),
- flange-to-flange width (rim width, inside dimension between the rim flanges),
- center-hole diameter (centering diameter, as size of fit),
- rim offset ET (dimension in mm from the rim center to the attachment face of the disk wheel),
- pitch-circle diameter (diameter of the circle on which the center points of the bolt holes are situated),
- flange height (measured from the rim nominal diameter to the saddle point of the flange radius).

The rim offset determines the position of the wheel in the vehicle. The wheel is generally secured to the brake-disk chamber or to the brake drum. With a rim offset $ET = 0$ wheel attachment is exactly in the middle of the rim width. Changing the rim offset changes the position of the wheel and with it the track width of the vehicle. A smaller rim offset equates to a wider

Figure 4: Hump shapes

- a) Hump,
b) Flat hump.
 D Nominal rim diameter,
 D_H Hump diameter.



track width, a larger rim offset equates to a narrower track width.

Rim size

The basic dimensions of a rim size using a commercial-vehicle disk wheel with 15° tapered bead seat rim by way of example is 22.5 × 8.25 inches. The first value gives the rim diameter in inches, the second value the flange-to-flange width in inches.

Rim designs

Depending on the intended purpose and the tire design, there are a range of different cross-sectional rim shapes available (Figure 6): Rims can be made from a sin-

gle part or from multiple parts. According to EU standardization single-part rims must be identified with a "x" (e.g. 6J × 15H2) and multiple-part rims with a "y".

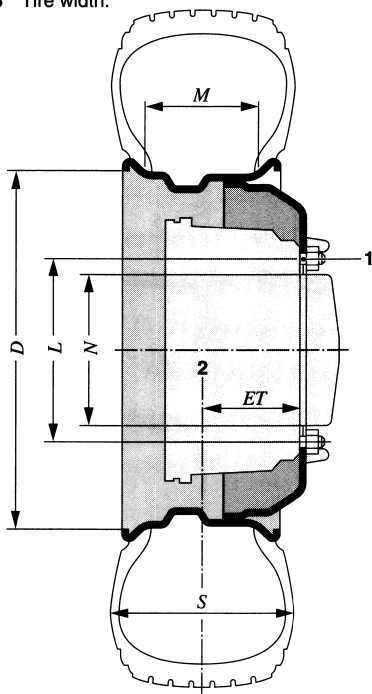
Drop-center rims for passenger cars and light utility vehicles

The rim base is recessed for tire mounting. The drop-center rim can be made

Figure 5: Rim and wheel dimensions

(Picture source: MAN Nutzfahrzeuge Group),

- 1 Bolt holes,
- 2 Rim center,
- D* Rim diameter,
- ET* Rim offset,
- L* Pitch-circle diameter,
- M* Flange-to-flange width,
- N* Center-hole diameter,
- S* Tire width.

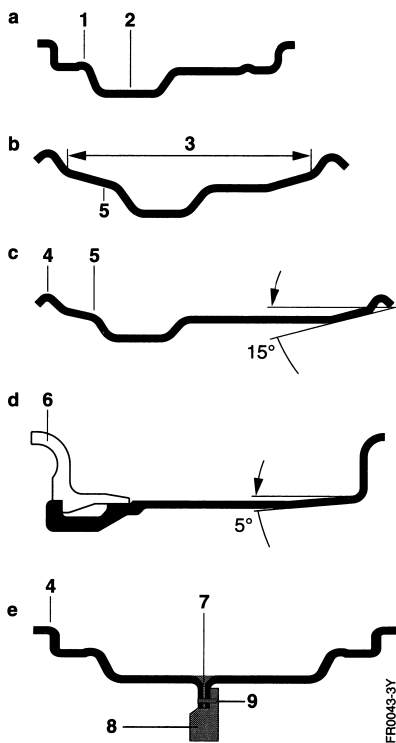


SFR0045-1Y

Figure 6: Rim design types

- a) Drop-center rim,
- b) 15° tapered bead seat rim,
- c) Wide 15° tapered bead seat rim,
- d) 5° tapered bead seat rim,
- e) Two-part passenger-car rim.

- 1 Hump,
- 2 Rim base,
- 3 Flange-to-flange width,
- 4 Rim flange,
- 5 15° tapered bead seat,
- 6 5° tapered bead seat,
- 7 Sealant,
- 8 Wheel disk,
- 9 Bolt.



SFR0043-3Y

from one part (Figure 6a) or two parts (Figure 6e). The two-part rim is divided into a front half and a rear half which are bolted to each other in the drop center through an all-round pitch circle and to the wheel disk. A sealing ring or a sealing compound is used to provide the seal. The two-part variant traces its origins back to motor sport and offered the advantage of being able to replace one rim half in the event of damage. The tire is mounted in exactly the same way as on the single-part rim.

15° tapered bead seat rim for commercial vehicles

The 15° tapered bead seat rim (Figures 6b and 6c) is made from a single part. The rim base is for tire-mounting purposes provided with a drop center to which the 15° tapered bead seats are connected. The 15° tapered bead seat rim with drop center is necessary to enable the benefits of the tubeless tire to be exploited on heavy commercial vehicles too.

5° tapered bead seat rim

The 5° tapered bead seat rim (Figure 6d), also called flat-base rim, is made from multiple parts. This is needed to mount the tires. The outer 5° tapered bead seat, which is non-permanently connected to the outer rim flange, can be removed. The outer 5° tapered bead seat is held on the rim base by an all-round sealing ring.

The tire is pushed onto the rim base when the 5° tapered bead seat is removed. The fact that the rim is made from multiple parts means that a tube is required. The two bead seats have a 5° taper.

These rim systems have advantages when it comes to changing tires. However, they are heavier than 15° tapered bead seat rims and do not exhibit the good radial and lateral true-running properties of single-part rims.

Special trims

for special wheel/tire systems

PAX (Pneu Accrochage, X stands for Michelin radial tire technology) and CTS (Conti Tire System) are distinct rim geometries which can only be used with specially developed tires. This design is used primarily on armored vehicles. The

complete system is designed to permit continued driving with a punctured tire without the tire coming off the rim. The two systems also prevent the tire from being destroyed by thermal load during driving when punctured. With a conventional rim base the sidewall folds up when the tire is punctured and friction occurs.

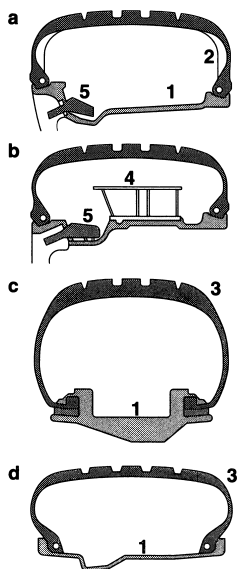
In the PAX system (Figures 3c and 7b) from Michelin an additional support ring is mounted on the rim base; the punctured tire rests on this support ring.

The CTS system from Continental does not have an additional support ring. The tire reaches round the rim base and can rest when punctured on the rim base (Figures 3d and 7c). Both run-flat systems are required by law to have a tire-pressure control system.

Figure 7: Special shapes for special wheel/tire systems

- a) PAX system with reinforced run-flat tire,
- b) PAX system with support ring,
- c) CT system (cast-wheel variant),
- d) Conventional wheel-tire system.

- 1 Rim,
- 2 Run-flat tire with reinforced sidewall,
- 3 Tire,
- 4 Support ring,
- 5 Valve with tire-pressure control.



Design criteria

Passenger-car wheels

Design criteria for passenger-car wheels include:

- High durability,
- Good support for brake cooling,
- Reliable wheel mounting,
- Low radial and lateral run-out,
- Small space requirement,
- Good corrosion protection,
- Low weight,
- Low costs,
- Problem-free tire fitting,
- Good tire seating,
- Good balance-weight seating (see Wheel with tire),
- Appealing design (for aluminum wheels),
- In part requirements to improve vehicle aerodynamics (c_d value).

Special requirements for commercial-vehicle wheels

Wheels for commercial vehicles are technologically highly sophisticated. Whereas racing-car wheels are geared towards maximum speed, commercial-vehicle wheels carry high tonnages at simultaneously – compared with excavators for example – high speed. In Europe, for example, long-distance commercial vehicles travel with a total weight of 40 tons at a speed of 80 km/h. It is essential when establishing the minimum tire size for a commercial-vehicle axle always to start out from the permissible axle load and the top speed dictated by the vehicle design.

The larger the load to be transported and the rougher the terrain to be negotiated, the more important it is to take into account the – sometimes competing – rim-design requirements. High load capacity is to be achieved by designing an appropriate wheel shape and using optimum materials. High fatigue strength is essential to increasing road safety. Low wheel weight optimizes payload and is required because the wheel, as an unsprung, rotating mass, influences the overall vehicle oscillatory system.

Designations for passenger-car wheels

A typical wheel designation for passenger-car wheels is, for example:

6 ½ J x 16 H2 ET30.

6 ½	Dimension of rim width in inches,
J	Rim-flange geometry,
x	Single-part rim base,
16	Dimension of rim diameter in inches,
H2	Rim hump at the inner and outer rim bead seats,
ET	Rim offset,
30	Dimension of rim offset in mm.

The designation and the associated dimensions with permitted tolerances have been prescribed as binding and standardized by worldwide recognized standards organizations such as ETRTO (European Tyre and Rim Technical Organization) or ISO (International Standards Organization) to harmonize rim and tire dimensions.

Materials for wheels

There are basically steel and light-alloy wheels. The material must always be specified in connection with the manufacturing technology. The following overview is intended to simplify systematization.

Classification of wheels

Steel wheel

A steel wheel consists of two parts, the rim and the wheel disk. Steel wheels are made from hot-rolled sheet steel in a forming process under rolling and bending conditions and joined by welding.

Light-alloy wheel

Light-alloy wheels are usually made of aluminum or magnesium alloys. They are produced using a variety of technologies. The aluminum wheel is produced as a cast wheel, forged wheel, sheet-metal wheel, or hybrid wheel. The magnesium wheel is only produced as a cast wheel.

The advantages of lightweight wheels are improved vibrational behavior, responsive suspension, reduced fuel consumption, and higher payloads. Light-alloy wheels are used in commercial-vehicle applications in particular in weight-sensitive transportation jobs. These jobs include the transportation of tanks and silos, where not exceeding the maximum transportation weight is of primary importance. In these situations the higher-cost light-alloy wheels usually pay for themselves within the first year of use.

Plastic wheel

Plastic wheels are made in an injection-molding process from mineral-fiber-reinforced polyamide and with metal inserts.

Materials

Sheet steel

In all the cheapest variant of passenger-car wheels is manufactured from hot-rolled and pickled sheet steel strip unrolled from the coil. The very good mechanical properties of this material provide for thin-walled wheel designed which are manufactured in highly automated, high-precision forming processes under bending conditions to finishing dimensions with close tolerances.

The continuous trend towards lightweight construction particularly since the CO₂ discussion has accelerated the use of high-strength, fine-grained structural steel. Its high tensile strength (600...750 N/mm²) and very good formability and weldability enable lightweight, and cost-effective wheels to be manufactured efficiently.

Further potential for weight savings can be opened up by the use of "tailored blanks" for rim production. Here the sheet thickness of the starting material is adapted to the stresses in the wheel, in the course of which material strips of sheets of different thicknesses are joined by laser welding to create a blank.

Aluminum sheet

Aluminum sheet as an alternative to sheet steel can be more easily formed and is also – albeit thanks to more expensive methods (MIG welding) – more readily weldable. The manufacturing overhead is greater relative to steel wheels and the material costs are comparatively high, which prevents a wider range of applications. The use of high-strength sheet steels has greatly reduced the original weight advantage of aluminum sheet with the result that a cost/benefit analysis comes out in favor of steel wheels.

Light-metal alloys

Light-metal alloys are based in the main on aluminum alloys and in rare cases (e.g. in motor sport) on magnesium alloys. In the case of aluminum wheels, a distinction is made between cast and forge alloys depending on the manufacturing method used.

Cast alloys

Aluminum cast wheels are manufactured in low-pressure diecasting from aluminum alloys. The cast blank is formed in a steel mold, which is filled with liquid molten mass and cooled under controlled conditions to solidify. Aluminum alloys with a silicon content of between 7 and 11% are used, depending on whether good castability or high strength is to be achieved.

Two alloys have proven successful. GK- AlSi11 is used for small wheels (up to 16 inches) with low wheel loads. The outstanding castability thanks to the high silicon content provides for highly efficient production with low reject rates due to casting defects. This alloy cannot be hardened by heat treatment. The wheels are therefore designed with greater wall thicknesses, which is reflected in slightly higher weights.

GK- AlSi7Mg is used for large wheels with a higher wheel load and for weight-optimized wheels. Adding 0.2...0.5% magnesium to the aluminum alloy increases the strength of the cast wheel as a result of subsequent heat treatment (solution heat treatment and aging at warm temperature). This advantage is exploited to meet high load requirements during vehicle operation with a minimum of material usage.

To ensure that the high requirements with regard to strength, tightness and ductility placed on these safety-related vehicle components are met, only pure primary aluminum is used as the starting material. Contamination of the alloy with iron would cause acicular structures to form, which in turn would weaken the mechanical properties (elongation at fracture and tensile strength). Contamination with copper would reduce the chemical stability.

Forge alloy

Aluminum forged wheels are used in passenger cars when light, weight-optimized wheels are required and the target weight cannot be achieved with cast wheels. The increased hardness of the aluminum wrought alloy caused by the forging process (increase in mechanical strength due to plastic deformation) enables the wheel to be designed with a thinner wall, which means that less material needs to be used and thus the wheel is lighter.

The starting material takes the form of round continuously cast bars of AlSi1Mg which are sawn into precisely "portioned" disks. These are subjected to a three- to four-stage forging process to create the visible side (design side, disk, spider) and a flow-forming process to create the rim. As well as being hardened by plastic deformation the material is modified by a heat-treatment process.

Magnesium alloys

Whereas magnesium alloys have been unable to establish themselves in volume production – due to higher production costs in view of special safety precautions (risk of fire during cutting) – they are used in individual cases for special-purpose vehicles and racing cars.

Plastics

The use of plastic as a material for wheels is still in the developmental stage due in particular to insufficient high-temperature strength and difficult wheel mounting and manufacture. In particular the insufficient impact strength and thermal load capability as well as the incalculable long-term properties mean that plastic currently appears to make little sense as a material for a safety component such as the disk wheel in automobile construction.

Manufacturing processes

Steel wheels

When the text refers to passenger-car and commercial-vehicle steel wheels, exclusively sheet-steel wheels are what are meant. Other manufacturing processes such as casting and forging are not used for this material to manufacture wheels.

Passenger-car steel wheels consists of two parts: the wheel disk and the rim. They are welded to each other at the end of the production process. Both the wheel disk and the rim are manufactured in volume production on fully linked, highly automated production lines by means of forming processes under bending conditions. Smaller quantities can be produced more economically on individual presses.

Manufacturing the wheel disk

To manufacture the wheel disk the material required is unrolled directly from the coil, straightened flat and fed to a large transfer press with a force of pressure of approx. 40,000 kN. The press is equipped with a nine- to eleven-stage follow-on composite tool, in which the sheet blank is automatically routed with each press stroke from station to station.

A square blank with rounded corners is punched in the first machining step. Then in three to four stages the wheel disk and the center-hole area are formed by deep-drawing and stamping operations. In the next two or three stations the ventilation holes are punched with wedge driving tools and then stamped on the tool outlet side. The stamping replaces deburring of the pointed cutting edges and reduces the wheel's susceptibility to cracking under load.

Finally, the wheel designations are stamped on the reverse side and the calotte- or cone-shaped wheel-stud attachment faces are formed. In the last stage the wheel disk is calibrated to the finishing dimension for joining with the rim.

Manufacturing the rim

The rim material is likewise unrolled from the sheet coil, straightened and cut to length. The strips are stacked and forwarded to the automated rim production line. Here the sheet strip is shaped in a bending machine between three cylinder rolls into a ring and joined at the junction point in a butt-seam welding machine. The upsetting flash created in the process is then planed away on the inside and outside and the weld seam is smoothed by rollers. The unavoidable upsetting flash also on the side edges is then deburred and rounded.

The final rim contour is created on roller-burnishing machines by three consecutive forming steps, each with three profile rollers. The contour of the tool rollers is transferred to the rim here.

Where necessary, flow-forming is used to adapt the wall thickness and material distribution in the rim profile to the loads involved, thus saving more weight.

In the next operation the valve bore is produced on a rotary table. In the first station a flat attachment face is struck, from which the bore is punched in the second cycle. The cutting edges are then rounded on both sides by stamping.

Finally, the rim is drawn in a press onto a gage with the exact finishing dimension in order to create the tight concentricity and lateral-running tolerances (calibration).

Joining the wheel disk and rim

The process of joining the wheel disk to the rim creates the steel wheel. The two wheel parts – wheel disk and rim – are routed to a fully automated welding installation. On a small press they are aligned to each other and then joined to each other in a precisely specified position.

In the next station the assembly is connected by shielded arc welding with four to eight weld seams. These are deslagged and cleaned by rotating brushes adapted to the wheel. Then the wheel is calibrated to the finishing dimensions and made available for surface treatment.

The manufacturing process for steel wheels is so precise that the blank wheel is able to move on to the painting plant without any mechanical reworking. There, as a general principle, it runs through cathodic dip coating and, if stipulated, is given an additional visually appealing finishing coat.

Sheet-steel wheels are characterized in particular by their robustness and, thanks to their low manufacturing costs, are offered in the entry-level equipment specs of vehicle models.

Aluminum wheels

Aluminum-sheet wheels

For the most part, the basic manufacturing process is identical to that of the sheet-steel wheel. In view of the lower strength, the wall thickness must be greater compared with the sheet-steel wheel. Despite engineers having full mastery of the technology, aluminum-sheet wheels have not gained acceptance because steel wheels as a whole are the more economical wheel variant and the design options are very limited.

Aluminum forged wheels

As the name suggests, a forged wheel is created by the hot forming of an aluminum round blank between two forming tools. The forming process takes place in two stages – the forging of the wheel front and rear sides between two defined tools and the flow-forming of the rim contour.

The starting material for aluminum forged wheels takes the form of 6 m long continuously cast bars of AlSi1Mg with a diameter of 200...300 mm, depending on the planned wheel size. After an ultrasonic test the cavity-free bar sections are sawn off to a predefined length. The saw support – a cylinder approx. 250 mm in diameter and 150 mm high – is fed to the automated forging line. This consists of a heating furnace and up to four consecutive forging presses with forces of pressure of 8,000...40,000 kN. Parts are handled between the presses by robots. The result of this first forming process is blank with the finished design of the wheel disk, a punched-out center hole and a ring-shaped material reservoir positioned around the circumference with the material scheduled for the rim.

In flow-forming operation involving three rollers the ring-shaped disk is split open (hence the term "split wheel") and rolled out on a bell-shaped tool to make the rim.

Before being machined, the forged blank is subjected to heat treatment to improve the mechanical properties. The complete wheel contour is turned on two consecutive lathes (i.e. machined to finishing dimension). Then the bolt holes and the valve hole are drilled and milled on a machining center. The concluding polishing process gives the wheel its reflecting luster.

This high-precision machining ensures that every wheel runs absolutely true. There is no radial or lateral run-out.

If small batch sizes do not justify the manufacture of a forging tool, a special form of forged-wheel production is used. This involves the manufacture of a forged blank in the shape of a thick-walled cylinder whose base is a disk representing the rotation contour of the design and of the inner side of the wheel. With high machining overhead, the wheel design is usually 100% milled and the rim base first flow-formed and then turned.

Aluminum cast wheels

The most common method used is low-pressure mold casting. In the casting machine the aluminum molten mass tempered under controlled conditions is located in a crucible underneath the mold. The mold and crucible are connected by means of a feed tube. Once the mold is closed, the pressure in the crucible is increased to approx. 1 bar, causing the molten mass in the feed tube to rise and fill the mold.

Fusion heat is drawn off during the solidification process through precisely defined cooling channels in the mold. The specific cooling and heat removal during the solidification process and the casting parameters in their entirety (pressure, temperature and time) are decisive factors for the casting quality.

The production process is completely automated from the casting stage. The cast blanks removed by robot arms pass via linked conveyors through the following machining stages until they are automatically stacked and packed as wheels in the dispatch area:

- casting,
- removal of riser bore,
- X-ray test,
- heat treatment,
- machining,
- brushing and deburring,
- leak test,
- painting,
- dispatch.

In the X-ray test all the blanks are tested for casting defects according to specifications stipulated by the customer. The defective parts with casting defects that are not visible from the outside, e.g. porosity or shrink holes (cavities, material breaks) and inclusions or contaminants, are separated out and returned to the melting furnace.

The automated run is interrupted only before the painting stage so as to form production batches which are painted the same color.

Flow-forming process

Where necessary, it is possible to use a modified, slightly more sophisticated process to manufacture weight-optimized cast wheels, so-called “flow-forming wheels”. This produces a weight saving of around 0.9 kg for a 19” wheel. For the flow-forming process, the cast blank is made in a similar way to that used in forging. A ring is provided around the design surface instead of the formed rim contour as a material deposit for the rim. This ring is processed in a specially designed machining cell as follows:

- preturning for rolling,
- heating,
- rolling out of the rim (flow-forming).

The blank formed in this way is returned to the “normal” manufacturing process prior to heat treatment.

Another way of manufacturing weight-optimized wheels is to insert lost cores in zones of the wheel subject to less stress. Thus, aluminum is replaced by cavities, for example in the spoke, but also much more rarely in the hump. Accordingly there are hollow-spoke wheels and hollow-hump wheels.

Squeeze-cast process

The squeeze cast process attempts to exploit the advantages of diecasting for aluminum wheels. An exactly portioned amount of molten aluminum is pressed under high pressure into a diecasting mold under exactly defined casting parameters. The great advantage lies in the high solidification speed with positive effects on the material structure. Other advantages are the significantly lower machining overhead – and thus less material usage – and the relatively high output and longer mold service life. This casting process is used in individual cases, but it requires special, relatively complex casting machines and molds. This process has yet to gain acceptance.

Wheel design variations

Single- or multiple-part design variation

Sheet-steel and aluminum-sheet wheels consist of two parts. In this design wheel disk and rim are welded to each other. In the case of forged light-alloy wheels and cast wheels, the one-piece version dominates.

Multiple-part versions, even those which are made of different materials (e.g. magnesium wheel disk and aluminum rim), are available mainly for tuning applications and for sports vehicles. Multiple-part wheels trace their origin back to motor sport. Mechanics exploited the advantage of being able to replace the damaged parts. In tuning applications, for example, the possibility of standardized rim rings and disks is used to create a large number of different wheel dimensions. For the most part, however, multiple parts no longer have a technical background and are now only used for visual reasons. Multiple-part wheels are subdivided into two- and three-part wheels.

Wheel spider

The wheel spider on cast wheels refers to the area of the spokes, which is represented in a steel wheel as the wheel disk and is for the most part provided with openings. Ventilation holes, slits and openings in the wheel disk or the wheel spider serve on the one hand to reduce weight, and on the other hand to ventilate the brake system and to enhance the visual design of the wheel. Set against this is the present-day requirement also to optimize the effects of the wheel with regard to the entire aerodynamics of the vehicle. Depending on the dynamic behavior of the body, a positive effect can be achieved here by keeping the area of the openings small and fashioning the geometry of the spokes as flat as possible. This measure has a more negative impact on the weight of the wheel such that it is now more common to find plastic parts on an aluminum wheel which are intended to aerodynamically optimize the wheel.

Rim variants

Rims used for passenger cars, vans and light utility vehicles are almost always drop-center rims with H2 double humps (rarely with FH or FH2 flat humps), tapered bead seats, and J section rim flange. Less common on smaller vehicles is the lower flange shape B, which today is used primarily in compact spare wheels. The higher flange shapes, JK and K, are rare on modern vehicles, and then only on heavy vehicles.

Lightweight-construction technologies

Hollow-spoke technology with sand casting cores or ceramic cores remaining (lost) in the wheel also provides good possibilities to reduce weight, but it requires a suitable design and special manufacturing equipment. This process is also associated with higher costs.

More widely used today is the flow-forming process for cast wheels, in which the rim base is only partly precast and then rolled out by machine to the corresponding rim width. The compressed material provides for thinner wall thicknesses with reduced weight in the rim base.

"Structural wheels" are used, among other things, as spare wheels or as road wheels with plastic wheel covers. Unrestricted by design conditions, the aim here is to use the minimum quantity of material possible to guarantee operating and functional safety, as well as streamline production costs for these wheels.

Wheel mounting

The design of the wheel and the mounting elements must meet safety requirements in all vehicle operating conditions. The wheel forces resulting from motive force, brakes, wheel load and wheel location must be supported by the overall mounting system (wheel bolt, wheel hub, brake-disk chamber, wheel-bolt holes, possibly coatings of parts) without impairing fatigue limits or the function of the wheel and axle components. Careful coordination of the friction parameters and geometry at the wheel bolts or the wheel nuts and contact zone of the wheel (bolt head to wheel-bolt hole) is essential when specifying the tightening torques in engineering design and in practice.

The geometric configuration of the wheel mounts in pitch diameter, number and dimensioning of the mounting elements are subject to the needs and requirements of each vehicle manufacturer. The wheel on a passenger car is secured to the axle hub by three to five wheel bolts or wheel nuts inserted through the mounting holes. Off-road vehicles and light utility vehicles often have six wheel bolts or wheel nuts. Commercial vehicles as a rule have ten wheel nuts, but this number can sometimes be even higher (e.g. tractors and excavators). The design of the contact surface of the nuts varies, depending on the vehicle manufacturer (e.g. calotte, cone, flat head). The longitudinal bolt forces that are decisive for the durability of the bolted connection must be reached and adhered to, both when new and used, in all dynamic operating states.

The high degree of true-running is achieved by means of a central wheel mount at the wheel hub with a precise alignment shoulder.

At present, wheels mounted with a central nut and interlocking driving pins are used almost exclusively on racing cars.

Wheel trims

Wheel trims (wheel caps) are mainly used for visual reasons on steel wheels and are affixed to the wheels using elastic retaining-spring elements which are easily detachable. But today aluminum cast wheels also feature wheel trims, which are often intended to improve aerodynamics. The design of aluminum wheels is for the most part kept simple and the weight kept low. Bolted solutions are also used in rare cases. The material used most often for wheel caps is heat-resistant plastic, e.g. polyamide 6. However, in some cases, aluminum and stainless pressed steel are also used.

Special wheel/tire systems*TRX rim*

More recent rim developments, which have produced in limited series, are the TR rim (in metric dimensions). They were developed by MICHELIN for use with matching TRX tires and provide more room for the brakes.

Rims made by DUNLOP with a Denloc groove also require special tires; at low tire pressure and also in the case of pressure loss, the system is supposed to prevent the tire from coming off the rim, enhancing safety and mobility.

The TD system (TRX-Denloc) brings together both wheel/tire systems. As opposed to common practice, the two above designs are have rim and tire matched to each other and are unable to combine rim and tire with other tire versions, or only to a very limited extent.

CTS and PAX systems

The CTS/CWS and PAX systems were able to dispense with the need for a spare wheel. The original idea behind using these two systems to save on the spare wheel failed to catch on in the market and today these are used primarily on armored vehicles (see Rim designs).

Compact spare wheels

For space-saving reasons a compact spare wheel (mini spare) is often used as the spare wheel. This can be stowed in combination with a collapsible spare tire in an even smaller space (e.g. in roadsters, convertibles). All compact spare wheel systems are equipped with a specially designed tire where the driving properties are only suitable for emergency operation and limited top speed (approx. 80 km/h). Its benefits are the subject of debate, but it is becoming increasingly popular compared with a full-size spare wheel.

In many countries it is no longer required by law to carry a spare wheel. Instead, vehicles are equipped with a puncture kit (Tire-Fit) to repair any tire damage. The puncture kit consists of an electrically driven compressor and a sealant, which is pumped into the tire through the valve.

Stress and testing of wheels

The extremely varied and complex stress conditions in the wheel as a component in conjunction with a wide variety of operating conditions in the vehicle require specific endurance tests in order to be able to confirm the durability of a wheel with acceptable overhead. In general, the dynamic tests are run in test laboratories on standardized testing units, whereby a simulation of road operation that is close to reality is simulated and a good correlation of the test results to pure road operation is achieved. Country-specific legal requirements make special tests necessary, e.g. in the case of light-alloy wheels the simulation of a side curb impact (impact test).

Testing of sheet-steel wheels

The critical zones on a sheet-steel wheel are in particular the zones around the weld seams, mounting boreholes, dish (curvature of the wheel disk, see Figure 1), and ventilation holes. The operating conditions in each case, for example straight-ahead driving and cornering, generate different damage patterns in the area of the welding seam on the drop center well and in the wheel disk. Tests of the material quality and of the welded joints as well as surface tests are backed up by the endurance tests and indicate the need to optimize wheel manufacturing.

Testing of light-alloy wheels

Light-alloy wheels run through a similar testing process whereby, unlike sheet-steel wheels, the more varied influencing parameters of material, manufacturing, and design mean that the test requirements are at a significantly higher level. This ensures that fluctuations in material and manufacturing are unable to lead to premature failure. The maximum stresses occur mainly on the back of the wheel in the supporting structure of the ribs and spokes, in rare cases on the visible side.

The material quality and processing have a great influence on the durability of aluminum cast wheels. Inadequate physical values such as elasticity (during expansion) and tensile strength can be caused by poor heat application during casting or during heat treatment. This leads to porosity and shrink holes and deficient structural formation. The burrs that occur in high-stress zones during machine-cutting represent preliminary damage similar to notches and are often the starting point for incipient cracks. Careful machine deburring of these zones or specific constructive countermeasures, e.g. generously molded radii, are essential.

Testing of wheel with tire

Concentricity and lateral running

To assess the concentricity (true running) quality of a wheel on a vehicle, it is necessary to assess the wheel with the tire fitted, i.e. as a wheel with tire. During the manufacture of a wheel for concentricity the hub centering is in proportion with the two areas for the inner and outer tire seats. Likewise the contact face on the wheel hub and the inner areas of the rim flanges are responsible for the lateral running of the wheel. As a result of production these areas are endowed with tolerances (0.3 mm is usually given for concentricity and lateral running for passenger-car wheels), with which the tolerances of the tire now overlap. This can positively or negatively influence the concentricity of the wheel with tire. "Matching" is used to facilitate optimum concentricity of a wheel

with tire. In this process the wheel and the tire are positioned during fitting in relation to each other in such a way that the "concentricity high point" of the wheel matches up with the "low point" of the tire.

On the wheel the high point is determined from the concentricity measurements of the two tire-seat areas. For each area an individual high point is obtained with different angular positions on the circumference of the wheel. These two values produce through vector addition a common value with a resulting angular position. This position is marked on the wheel with a colored dot or an adhesive dot.

On the tire the low point corresponds to the position where it reaches the lowest force variation while rolling. It is also marked with a colored dot. From a technical viewpoint the tire can also be compared with a spring that exhibits a radial stiffness. Based on production the tire can never be manufactured to such precision that it exhibits a uniform stiffness over its entire circumference. Wheels with tires with poor concentricity make themselves felt on the vehicle not only through a radial movement of the body (i.e. in the z direction). In the direction of travel a minimal variable force from acceleration and braking is also experienced with each wheel rotation.

Good wheel centering is extremely important for commercial vehicles that travel at higher speeds, but also when the wheels in question are large and heavy. On commercial vehicles that travel at higher speeds in particular, the lowest possible radial and lateral run-out on both rim bead seats and flanges is essential to ensuring smooth running. This increases safety and fuel economy.

Imbalance

Just as important as concentricity and lateral running to a smooth-rolling wheel with tire is compensation of the differently distributed masses on the wheel and tire. To this end it is necessary to minimize the influences of the masses on the rotating

wheel with balance weights by balancing. Normally passenger-car wheels are dynamically balanced due to the rim width, i.e. measurements are taken on two planes (inner and outer tire seats) and the required compensation mass is determined. This is then applied with balance weights at the point indicated by the balancing machine. Adhesive, clip-on or drive-on balance weights are used for this purpose (Figure 8). The ideal position for the balance weights on the wheel for dynamic balancing is the maximum distance to the rim center at as large a diameter as possible.

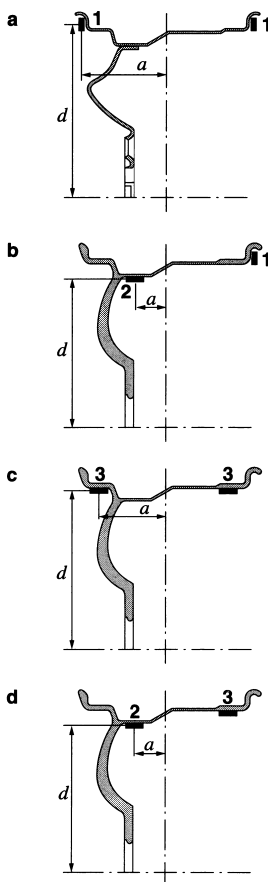
An imbalance, referred to as "residual imbalance", of 5 g per balance plane depending on the vehicle type and suspension is not discernible in the majority of vehicles. For each balance plane a balance weight should be attached at one position only. If a relatively high balance mass (over 80 g) is required in a balance plane, it is recommended to turn the tire on the wheel and repeat the balancing process. The lower the balance-weight mass on the wheel, the lower the residual-imbalance potential also.

Narrow wheels for two-wheeled vehicles are balanced on one plane only (with a balance weight in the rim center). This method is called static balancing.

Wheels for commercial vehicles and compact spare wheels with a restricted top speed are not balanced.

Figure 8: Balance-weight positions

- a) Clip-on or drive-on weight on the inside and outside (visible),
 - b) Clip-on or drive-on weight inside in combination with an adhesive weight under the drop center (concealed),
 - b) Clip-on or drive-on weight inside in combination with an adhesive weight under the tire seat (concealed),
 - d) Two clip-on drive-on weights (concealed).
 - 1 Balance weight with retaining spring clipped onto the rim flange,
 - 2 Balance weight glued to the inside of the wheel drop center,
 - 3 Balance weight glued to the inside of the rim bead seat.
- a Distance of balance weight to wheel center,
D Distance of balance weight to axis of rotation.



Tires

Function and requirements

The tire is the only component of a vehicle that comes into contact with the road. It thus assumes a key driving-dynamics position. Downstream driving-dynamics control systems such as the Antilock Braking System, the Traction Control System and the Electronic Stability Program are only ever as effective as the tire allows within the framework of its instantaneous power-transmission potential. When it comes to the tire, ultimately the crucial factor is the active safety of the vehicle.

Tires perform a variety of functions in everyday driving applications: They cushion, damp, steer, brake, accelerate, and simultaneously transmit forces in all three dimensions – at high and low temperatures, in the wet, on dry roads, on snow, mud and ice, on asphalt, concrete and pebble stones. They are meant to roll straight, permit precise steering, absorb road irregularities, bring the vehicle safely to a stop, and be quiet and comfortable. They are also meant to last, retain their characteristics with increasing age and decreasing tread depth, and produce as little rolling resistance as possible. Furthermore, an inflated tire performs supporting, vibration-damping and comfort-giving

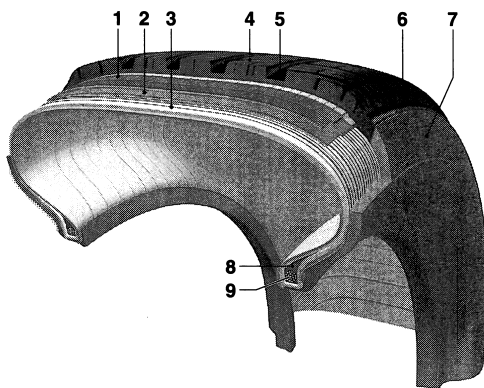
functions, and is therefore an active and fully integrated suspension element. The requirements resulting from these functions to be met by tires can be summarized as follows:

- high-speed resistance,
- durability,
- abrasion resistance (mileage),
- low rolling resistance,
- good properties in the wet (aquaplaning, wet braking, wet handling),
- good tire comfort, quiet rolling noise,
- good running characteristics in the limit range,
- resistance to aging,
- precise steering properties (handling),
- short braking distances,
- simple installation and assembly,
- true running and uniformity,
- economy,
- resistance to damage,
- resistance to chemicals.

The fundamental and visible material of a tire is rubber, an elastic to viscous material to which a tire owes the majority of its typical properties that are so important to the vehicle.

Figure 1: Tire construction

- 1 Nylon binding,
- 2 Steel-belt assembly,
- 3 Radial textile cord plies (casing),
- 4 Tire tread (tread rib),
- 5 Tread (tread groove),
- 6 Tire shoulder,
- 7 Sidewall,
- 8 Bead apex,
- 9 Bead with bead core (steel core, numerous thin steel cables twisted to other).



Tire construction

Design and components

Tires are a complex construction of different, mutually influencing raw materials, components and chemicals. A standard passenger-car tire consists of up to 25 different components and up to 12 different rubber compounds.

Today, only the tubeless, radial-ply steel-belted tire constructed in two stages satisfies the stringent demands imposed by the automotive industry and consumers.

Ingredients

The ingredients of a radial tire are:

- natural and synthetic rubber (approx. 40%),
- fillers such as e.g. soot, silica, silane, carbon, and chalk (approx. 30%),
- strength members such as e.g. steel, aramide, polyester, rayon, and nylon (approx. 15%),
- softeners, e.g. oils and resins (approx. 6%),
- vulcanization accelerators, e.g. sulfur, zinc oxide, stearin (approx. 6%),
- anti-aging agents, e.g. UV and ozone blockers (approx. 2%).

Since 2010 the toxicologically serious softeners and paraffined oils have been subject to particularly stringent limits values in the EU. For this reason manufacturers are increasingly using uncritical natural oils (e.g. sunflower oil).

Casing

The casing is stretched over a thin inner liner of airtight butyl rubber (Figure 1). Around 1,400 rubberized cords of rayon, nylon or polyester are combined in one or more casing plies to form the decisive strength member, the elastic “shell” of the tire. The cords run radially, i.e. at right angles to the tire plane from bead to bead – hence the designation radial tire. Cross-ply tires, in which casing cords are placed diagonally to the tire plane, in practical terms no longer play a role in modern-day applications.

Bead

The bead performs the important function of ensuring that the tire is securely and tightly seated on the rim. Driving and braking torques are transmitted via this crucial connecting point from the rim to the tire tread and thus to the road surface. Seated in the bead core is a cable of numerous steel wires, each of which can bear a load of up to 1,800 kg [1].

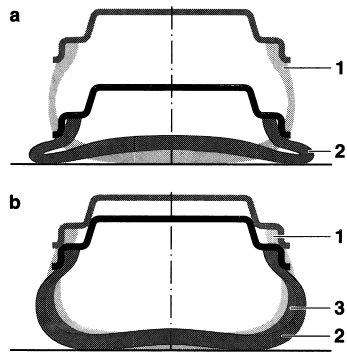
Sidewall

A thin and highly flexible rubber flank forms the sidewall and thus the flexible zone of the tire. The sidewall (tire flank) is however also the area of the tire that is most sensitive to damage.

Trouble-proof run-flat tires on the other hand have much thicker sidewalls than conventional designs (Figure 2). In the event of a blow-out the rim does not slump onto the tire casing and thus cannot damage it. In addition, fully deflated run-flat tires ensure a certain degree of steerability and directional stability for a further 80 km at speeds of up to 80 km/h.

Figure 2: Comparison of run-flat tire with standard tire

- a) Standard tire,
b) Run-flat tire.
1 Tire with normal inflation pressure,
2 Tire without inflation pressure,
3 Reinforced sidewall.



Cambering to blank

The finished, cylindrical combination of casing, inner liner, bead and sidewalls is pushed over a drum, the outside diameter of which corresponds to the inside diameter of this preliminary tire stage and to that of the subsequent tire. On this drum the cylindrical combination is cambered into the "real" tire shape (inflated and fixed) and then built further.

Because the casing cords run radially, i.e. transversely, to the rolling direction, the casing on its own would not be able sufficiently to transmit lateral forces when cornering and peripheral forces when accelerating and braking. It therefore needs support. This job is performed by the steel-belt assembly placed on top. Two or more plies of twisted and brass- and rubber-coated steel wires (steel cord) run not in the peripheral direction, but alter-

nately at acute angles of between 16° and 30° to each other. High-speed-resistant tires are additionally stabilized by a nylon or aramide binding which suppresses a peripheral increase caused by centrifugal force. The tread surrounds the casing.

From blank to finished tire

The tires in this penultimate stage is called a blank and is now placed in a heating press. Inside this press is an exchangeable recess – an exactly shaped negative mold of the later finished tire. In this heating mold the tire blank is "baked" under steam pressure (approx. 15 bar) and heat (to 180 °C) for up to 30 minutes and acquires its final, typical appearance. The tread rubber creeps on heating exactly and without cavities into the tire negative mold of the heating press, thus creating the tread pattern and the sidewall markings. As a result of sulfur added in a previous process the hitherto plastic rubber vulcanizes into elastic rubber and acquires its desired operating characteristics.

The tread pattern ensures low rolling resistance, water expulsion, good grip, and high mileage.

A finished tire of the standard size for the medium-size car class 205/55 R 16 91 H weighs around 8.5 kg. A commercial-vehicle tire of the standard size 385/65 R22.5 weighs around 75 kg.

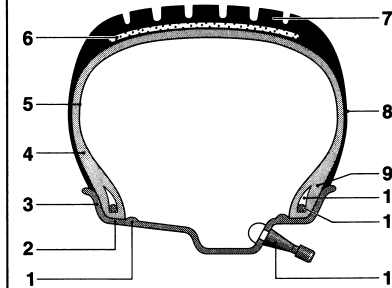
Tire with wheel

Together with the rim, the tire valve and the balance weights the tire forms the car's operational wheel (Figure 3). The rubber-elastic tire is inflated with compressed air, at which point it is able to absorb and transmit forces. Tire inflation pressure is usually 2...3.5 bar for a passenger car and 5...9 bar for a commercial vehicle. It is not the tire itself but rather the inflation air that supports the weight of the vehicle.

Figure 3: Design of rim with tire

- 1 Hump, 2 Rim bead seat,
- 3 Rim flange,
- 4 Casing (cord carcass),
- 5 Airtight rubber liner,
- 6 Steel belt,
- 7 Tread,
- 8 Sidewall,
- 9 Bead (with bead foot, bead core and bead apex),
- 10 Bead apex,
- 11 Bead core with steel core,
- 12 Valve.

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Differences between commercial-vehicle and passenger-car tires

Commercial-vehicle tires and generally similar in design to passenger-car tires, but are bigger, wider and heavier. Tire inflation pressure at 5...9 bar is much higher than approx. 2...3.5 bar for passenger-car tires.

The primary development objective is, as with passenger-car tires, to have all the parameters in proper proportion and above all the mileage. Commercial-vehicle tires therefore have comparatively hard, low-wear treads which are also regroovable and retreadable. Retreading of a bald tire identified as such is possible if the tire casing is undamaged.

Although the rolling resistance of a truck tire is lower than that of a passenger-car tire, its influence on the truck fuel consumption is greater due to the higher vehicle weight and the number of axles. In addition to high load capacity (up to 3...4 t per tire), other important tire characteristics include good straight-running stability, good cornering stability and traction. The trend in truck tires is towards increasingly smaller tire dimensions. This increases the useful load height and thus the transport volume.

Tire inflation pressure

Motor-vehicle manufacturers specify two values for tire inflation pressure for every vehicle – the part-load air pressure for a partly laden vehicle and the full-load air pressure for a fully laden vehicle or high driving speeds. The values are based first and foremost on the vehicle weight, on its top speed, on the tire design, and on the tire size. As a rule, the front and rear axles are subject to different desired pressures. The tire inflation pressure may only be measured and adjusted on a tire that is cold, i.e. that has not been heated up by vehicle operation.

A correct tire inflation pressure is important for

- optimum tire contact patch and optimum ground contact,
- shortest possible braking distance,
- optimum wet grip,
- balanced cornering stability,
- low rolling noises,
- low rolling resistance,
- low flexing work and heat production.

An excessively low tire inflation pressure will result in each case in the opposite end of the above parameters and in

- reduced service life,
- increased and sometimes uneven abrasion,
- progressive structural damage,
- danger of sudden tire blow-out,
- increased risk of accident,
- increased fuel consumption.

An excessively high tire inflation pressure (much less critical compared with low pressure)

- results in impaired tire comfort,
- reduces the ground contact patch (tire “stands”) and thus diminishes cornering-stability and braking-force potential,
- causes increased central abrasion,
- but only reduces the rolling resistance slightly.

Tire tread

A tire is provided over its entire circumference with geometrically shaped tread grooves, ribs and channels as well as additional notches forming gripping edges (sipes) (Figure 4).

The most important function of the integrated tread (not cut, but heated) is to adequately absorb and disperse water on the road surface (also snow and mud in the case of winter tires), since wet and even surface moisture alone have a negative effect on grip characteristics. The braking distance is therefore dependent not only on the effects of interlocking and adhesion produced by the two friction partners tread rubber and road surface (see Tire grip). The braking distance on wet roads increases as tire wear increases.

Minimum tread depth

Summer tires

The minimum tread depth prescribed by law (EU Directive 89/459 of 1989 [2]) is established in most European countries at 1.6 mm (for passenger cars).

Winter tires

The minimum tread depth for winter tires varies greatly from country to country. In Austria, for example, winter tires for passenger cars are required to have a minimum tread depth of 4.0 mm.

Wear-detection aid

The main tread-groove base features several rubber bumps exactly 1.6 mm high spread across the entire tread for the purpose of checking the minimum tread depth prescribed by law of 1.6 mm. If such a wear marker (TWI, tread wear indicator) on the tire makes contact with the road surface, the tire may no longer be used in road traffic.

Truck tires bearing the wording "Regroovable" on their sidewalls may be regrooved by the regrooving depth approved by the tire manufacturer (depending on the tire version 2...4 mm), ideally when the remaining tread depth is still 2 to 4 mm. Essentially, passenger-car tires are not permitted to be regrooved.

Aquaplaning

At higher speeds or when there is an enclosed layer of water on the road the tread the tread is no longer able to absorb sufficient water in itself and disperse it to the sides and to the rear. A wedge of water forces its way between the tires and the road, the tires lose contact with the road, and the vehicle loses its controllability – aquaplaning occurs (Figure 5).

Figure 4: Tire tread

- a) Typical tread of a summer tire,
 - b) Typical tread of a winter tire.
- 1 Wear marker.

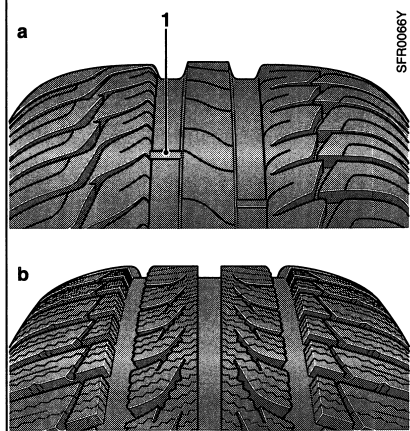
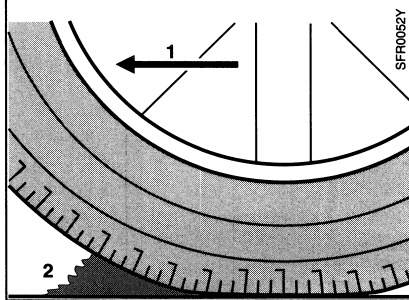


Figure 5: Aquaplaning

- 1 Driving direction,
- 2 Splash area.



The tire loses contact at precisely the stage when the pressure of the wedge-shaped splash water in front of the tire exceeds the pressure of the tire on the road. This pressure is squared as the driving speed increases. Because the critical pressure at which the tire aquaplanes is approximately equal to the internal tire pressure, passenger-car tires with an inflation pressure of approx. 2.3 bar aquaplane at a much lower speed than truck tires with 8 bar. Driving with a lower inflation pressure than prescribed decreases the already low speed from which aquaplaning occurs, once again significantly in passenger-car tires.

Tire contour, tread design and tread depth can defer the speed from which the risk of aquaplaning arises. Narrow tires, due to the higher pressure on the road (surface pressure, weight per contact patch), essentially aquaplane at higher speeds than wide tires, and they also heave to channel a much smaller volume of water. Drainage channels and rounded contact surfaces on wide tires reduce their risk of aquaplaning to an acceptable level. To compare: A tire with a nominal width of 220 mm must at 80 km/h and a rainwater height of 3 mm disperse

roughly 15 liters per second so as not to aquaplane. This figure is 10 liters for a 140 mm "narrow" tire.

Wet braking

Figure 6 shows the further point at which the tread depth is of crucial importance to road safety: The braking distance on a road wet with rain is roughly 50% longer with an almost bald tire (tread depth 1.6 mm) than with a new tire of identical size (tread depth 8 mm).

The residual speed v_R denotes the speed of the worse braking vehicle at the moment when the vehicle with better tires comes to a stop. It is calculated as

$$v_R = \sqrt{v_0^2 \cdot \left(1 - \frac{s_1}{s_2}\right)} \text{ in m/s,}$$

where

v_0 Driving speed at start of braking in m/s,

s_1 Braking distance with vehicle 1 (with better tires) in m,

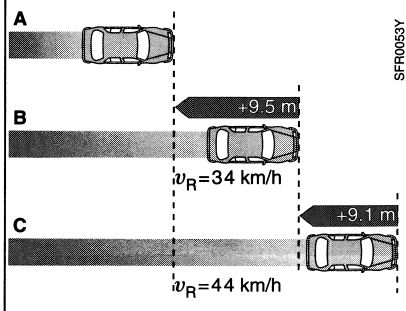
s_2 Braking distance with vehicle 2 (with bald tires) in m.

The residual speed is a measure of the theoretical accident severity to be expected in the event of a collision.

These calculated differences however can in practice only be effected by highly experienced drivers with extremely fast reactions. Many car drivers are not accustomed to ABS full braking; the total braking distance is lengthened significantly. Potentially shorter-braking tires only get a look-in if the vehicle is fitted with a brake assistant.

Figure 6: Braking distance from 80 km/h to a stop on a wet road with new tires and with bald tires [3]

- A Braking distance with 8 mm tread depth: 42.3 m.
- B Braking distance with 3 mm tread depth: 51.8 m.
- C Braking distance with 1.6 mm tread depth: 60.9 m.
- v_R Residual speed.



Force transmission

The four tire contact patches (footprint) are the direct and sole interface between the road surface and the vehicle.

Slip angle and slip

Only the tires rolling under an angle to the wheel rolling plane (slip angle α , Figure 7) and in the process deforming and constantly more or less slipping (see slip) transmit simultaneously and within physical limits the forces requested by the driver through steering, braking and accelerating. Conversely: A tire that does not roll or slip at an angle does not transmit any forces.

The tire transmits ever higher forces as the slip angle and slip increase. This relationship however is not linear. The effect is reversed after a relevant maximum value is reached (Figure 8). For passenger-car tires this reversal point for the slip angle is roughly $4...7^\circ$, corresponding to a pro-

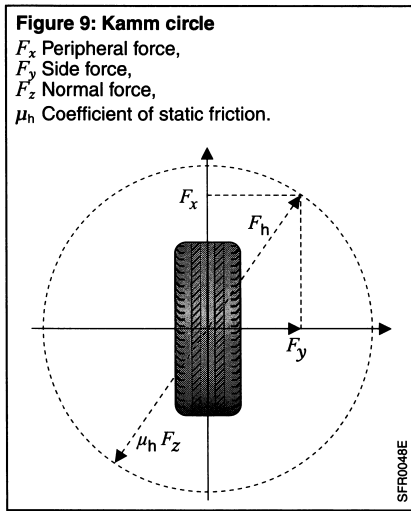
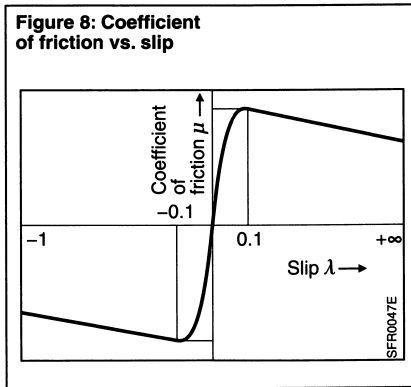
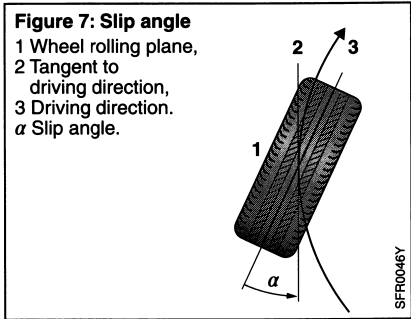
nounced steering-wheel angle. The reversal point for slip is $10...15\%$ (on snow up to 30%). Excessive slip is a consequence of excessive acceleration or excessively heavy braking. If the steering angle or brake pressure is increased further, the wheels lock if the vehicle is not fitted with an antilock braking system. The slip is then $\sim 100\%$.

Longitudinal and lateral forces

If a force F_x in the peripheral direction and a side force F_y occur simultaneously (e.g. when braking while cornering), the resulting transmitted horizontal force

$$F_h = \sqrt{F_x^2 + F_y^2}$$

cannot exceed the value $\mu_h F_z$. This situation can be explained by reference to the Kamm (friction) circle (Figure 9). The radius of the Kamm circle is equal to the maximum horizontal force $\mu_h F_z$ that can be transmitted via the tire. The maximum side force F_y is therefore smaller is at the same time a force F_x occurs in the peripheral direction. With the forces F_x and F_y marked in Figure 9 the wheel is exactly at the limit of the maximum horizontal force that can be transmitted.



Tire grip

Generation of grip

Tires must transmit all the dynamic forces to only four roughly postcard-sized areas. The grip required for this purpose between the tire contact patches and the road surface is generated by several simultaneously occurring phenomena. These are essentially positive locking (also referred to in this context as the interlocking effect) and adhesion by molecular forces of attraction.

When a vehicle driving past at a constant speed is considered from the outside, as the tire rolls the continuously changing ground contact patch remains apparently fixed in relation to the vehicle (Figure 10) – while each individual rubber block of the tire tread runs into this forcibly flattening contact patch, deforms, and is “ejected” again at the other end. Relative movements and thus slip are generated in this contact patch: Each individual rubber

Figure 10: Flattening in the contact patch between tire and road surface

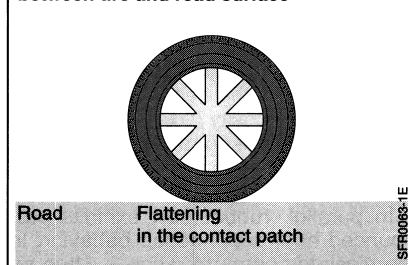
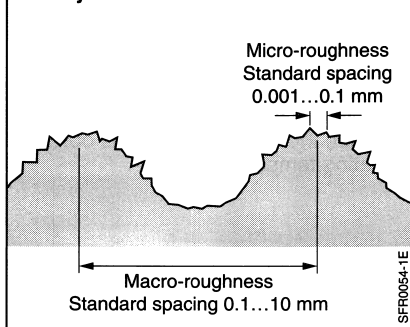


Figure 11: Micro- and macro-roughness of a dry road surface



block more or less slips during the dwell time in the contact patch.

Viscoelasticity

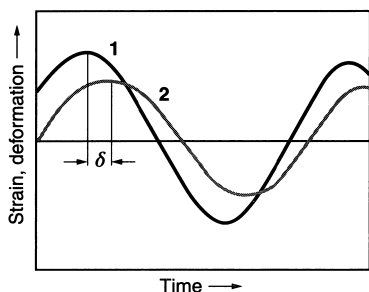
Viscoelasticity describes the time-, temperature- and frequency-dependent elasticity and the viscosity of polymer and elastomer substances (e.g. of plastics, rubber). Internal damping, molecular interlocking and creeping processes prevent the two extreme states “fully elastic” (e.g. like an elastic spring) and “highly viscous” (like a solid body). Deformation and the force that causes it as well as the mechanical strain pass off at different times.

Interlocking effect

The interlocking effect is created by the direct and intensive contact of the tire with the road, depending on the micro- and macro-roughness of the road surface (Figure 11). In the passage of the contact patch the tread block under consideration comes up against a bump in the asphalt, is upset, and slips off again on the other side of the bump at accelerated speed. Only when it produces slip in the process can it build up in the tangential direction a counterforce contrary to the rolling direction which counteracts the sliding and thus permits the transmission of steering, drive or braking forces.

Figure 12: Load application and load removal of a viscoelastic substance

- 1 Strain, force per unit of area,
 - 2 Deformation, elongation or upsetting relative to output variable.
- δ Phase lag.



On account of its viscoelastic properties a rubber block does not revert immediately to its original shape after deformation; the strain follows on from the deformation that causes it (Figure 12). This effect typical of rubber of hysteresis results, on account of the cyclic deformation of the viscoelastic rubber, in a loss of energy in the form of non-utilizable heat and thus in a contribution to friction (hysteresis friction). The component of the friction force parallel to the road surface thus facilitates the transmission of drive or braking forces.

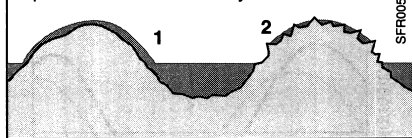
The principle of positive-locking grip also functions on damp and wet micro- and macro-rough road surfaces, but with limited effectiveness. Figure 13 shows that microfine asphalt points can penetrate the layer of moisture and thus the interlocking effect is maintained. However, an enclosed layer of water forms over the more rounded local bumps.

Essential to the interlocking effect is the presence of micro- and macroscopically small road irregularities. On a completely smooth surface (the coefficient of friction μ approaches zero) there would be no interlocking effect at all.

The interlocking effect and inner-molecular friction cause the tire to heat up. The resulting loss of energy is jointly responsible for the tire's rolling resistance, which makes up around 20...25% of a vehicle's fuel consumption.

Figure 13: Micro- and macro-roughness on a damp or wet road surface

- 1 Macro-roughness channels and stores the water, but cannot penetrate the water layer.
- 2 Micro-roughness generates local pressure peaks and can thus penetrate the residual layer.



Adhesion

Molecular adhesion is created by the interaction and intensive contact between tire and dry road. The creation and subsequent breaking up of adhesive connections at the contact points result in a contribution to the coefficient of friction (adhesion friction). On a wet road molecular adhesion fails while the interlocking effect remains effective.

The frequency range of molecular adhesion, excited by the micro-rough road surface during drastic braking and extreme cornering, encompasses the spectrum of $10^6 \dots 10^9$ Hz.

Load frequency and temperature

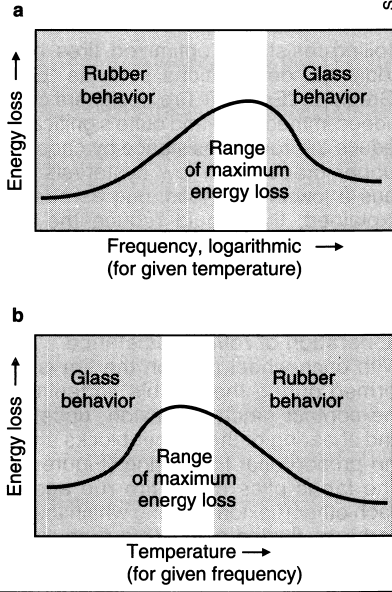
Two further important influencing factors on the quality of grip must be mentioned: If the tire when rolling is only excited with a low frequency (exposure frequency), the rubber behaves elastically (low energy loss, Figure 14a). In this low frequency range the rolling resistance is very low, the tire is relatively cold, and the grip is weaker. If, on the other hand, the frequency increases due to the excitations caused by micro- and macro-roughness, a viscoelastic behavior is manifested – the ideal range for tire grip (energy loss has its maximum). If the frequency continues to increase, viscosity (flowability) and energy loss decrease again, the is barely able to be deformed and hardens (glass behavior).

In parallel, rubber part exhibits pronounced heat-dependence behavior: In the glass-temperature range – when for example winter ambient temperatures cause the rubber compound to harden and become brittle (hence the analogy to glass) – the coefficient of friction of the tire rubber and with it the energy loss decrease markedly as a result of the rubber's molecular immobility (Figure 14b). Conversely, higher forces can be transmitted when the tire is moved in its optimum operating-temperature range.

Figure 14: Energy loss in the tire as a function of frequency and temperature

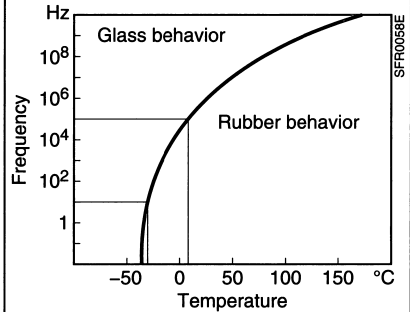
a) Behavior as a function of frequency,
b) Behavior as a function of temperature.

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For rubber, therefore, an inversely proportional dependence between temperature increase and exposure frequency can be ascertained. Thus the glass temperature of an elastomer rises from -20°C at only 10 Hz to $+10^{\circ}\text{C}$ at 10^5 Hz (Figure 15). Compound developers are able to design rubber compounds with a glass temperature of between -60°C and 0°C at a frequency of 10 Hz.

Figure 15: Rubber behavior as a function of frequency and temperature



Special case: Rubber friction

Coulomb's classic law of friction ($F_R = \mu mg$) does not apply to vulcanized tire rubber. Its coefficient of friction μ (also called friction value or friction factor) is not constant, but

- increases with decreasing surface pressure (force per contact patch, given in N/mm^2),
- decreases with increasing surface pressure,
- depends on the sliding velocity,
- depends on the temperature and exposure frequency.

The phenomenon of increased grip with decreasing surface pressure explains the importance of very wide (and thus low-surface-pressure) tires in motor sport.

Rolling resistance

Definition of terminology

Tractive resistances inhibit the forward movement of the vehicle and must be overcome by motive means. In addition to the aerodynamic drag, the frictional resistances in moving engine, transmission and chassis/suspension components, the climbing resistance, and inertia forces, the rolling resistance of tires is classed as one of the main tractive resistances. It contributes to approx. 20% of the fuel consumption on expressways/interstates, approx. 25% on orbital roads and approx. 30% on urban and ordinary roads [4].

Rolling-resistance decreases thus result directly in consumption and emission reductions. Rolling resistance (RR) corresponds to the energy loss per unit of distance and is given like every force in N (newtons). The dimensionless rolling-resistance coefficient c_{RR} denotes the ratio of rolling-resistance force to vehicle weight.

Example: Assuming a rolling-resistance force $F_{RR} = 120$ N and a vehicle weight $G = 10,000$ N ($G = mg$; vehicle mass m in kg, gravitational acceleration $g \approx 9.81$ m/s²) c_{RR} amounts to $0.012 = 1,2\%$. Standard values for c_{RR} for passenger-car tires on asphalt amount to 0.006 to 0.012, the (lower) value for truck tires to 0.004 to 0.008.

The "unit" kg/t (kilogram per ton) is occasionally used. In the example above the result 0.012 is 12 kg/t. This means in the case of a wheel load of 1 t = 1,000 kg that the rolling-resistance force F_{RR} assumes a value of 120 N.

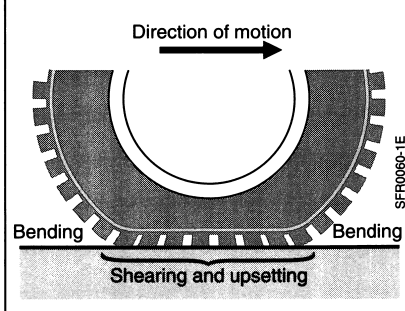
Rolling-resistance-optimized tires have additional designations such as "Eco", "Green" or "Energy". Tire designers could indeed immediately and quite significantly reduce the rolling resistance by choosing rubber grades with low hysteresis and thus a low energy loss, but, as already explained, this would reduce the grip values to unacceptable levels. In other words, lower fuel consumption comes at the expense of less grip.

Generation of rolling resistance

With each wheel rotation the tire is deformed during the forcible flattening in the contact patch by flexion, upsetting and shearing of the rubber blocks and of the proportional tire casing (Figure 16). The fabric plies of the tire rub against each other (flexion), during which the tire performs flexing work. This produces a viscoelastically caused energy loss in the form of non-utilizable heat. This heat loss makes up 90% of the rolling resistance.

Narrow tires and higher tire inflation pressure do increase the rolling resistance because contact patch and flexing work are reduced. However, designers are subject to very tight limits in terms of their scope of action in that the catalog of requirements with regard to handling performance, grip level and comfort are in direct conflict. Nevertheless, the technical specifications of the tire industry for future tire generations already feature tire dimensions such as 115/65 R 15 for subcompact-size cars and 205/50 R 21 for medium-size cars. Further drastic rolling-resistance reductions are not achievable with the requirements still standard today with regard to vehicle size, weight, maximum speed, sportiness, and comfort.

Figure 16: Shearing, upsetting and flexion in the contact lead-in



Even increasing the tire inflation pressure by 1 bar above the recommended value will only deliver a rolling-resistance reduction of 15%. In a vehicle with an assumed fuel consumption of 10 l/100 km this would only produce a saving of 1.6%. But because drivers still neglect to check tire pressure, the rolling-resistance reductions achieved in the most recent development cycles are not always implemented in real driving conditions. In addition, there is a direct conflict of aims between rolling-resistance optimization and the wet grip essential to road safety.

Conflict of aims between rolling resistance and grip

Frequency ranges

The deformations of the tread and rubber blocks caused by micro- and macro-roughness in the contact area between road surface and tire surface which generate the grip potential due to viscoelasticity occur in the very high frequency range of $10^3 \dots 10^{10}$ Hz. The energy loss caused by hysteresis is high here, resulting in the creation of high grip values.

However, the frequency spectrum which is important to rolling resistance is much lower at 1...100 Hz – precisely the range in which the inner tire structure is excited with each wheel rotation. At a driving speed of 100 km/h a passenger-car tire is deformed roughly 15 times per second, equating to a load frequency of 15 Hz. Tire designers nonetheless refer here to low-frequency excitation of the tire structure, particularly the casing.

Optimization incompatibility

These greatly differing frequency ranges explain the basic optimization incompatibility of simultaneously high values for grip and low rolling resistance. In conventional tires using industrial soot as the primary filler (up until the mid-1990s) a rubber compound with high hysteresis in the high-frequency grip range automatically results in a high energy loss in the tire components subjected to low-frequency load and thus in high rolling resistance.

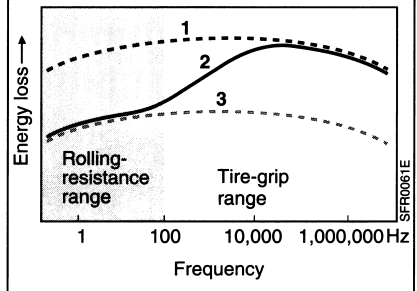
Silica to resolve the conflict of aims

The solution in the late 1990s involved the introduction of silica (trade name for refined silicate) as a gray-powdered filler, which increasingly replaced the standard industry soot used up that point. Together with auxiliary binding materials, or silanes, the conflict of interests between rolling resistance, grip and abrasion resistance can be elevated to a high level of compromise.

Silica-based rubber compounds exhibit low energy losses in the low-frequency range relevant to rolling resistance, but high energy losses in the high-frequency rubber-grip range (Figure 17). The curve for energy absorption has this rise steeply and thus advance in frequencies of $10^2 \dots 10^4$ Hz. This results in tires which have low rolling resistance but nevertheless very good grip.

Figure 17: Frequency dependence of energy loss

- 1 Rubber compound with distinct hysteresis (high tire-grip values),
- 2 Latest-generation rubber compound (combines low rolling resistance, good grip and high abrasion resistance),
- 3 Rubber compound with weak hysteresis (low rolling resistance, poor grip).



Rolling resistance and air pressure

Reduced tire pressure means increased flexing work and thus higher rolling resistance and impaired steering precision and braking stability.

Safety checks carried out in 2009 by the tire industry (Goodyear, Dunlop, Fulda) on 52,400 vehicles in 15 EU countries established that 81 % of all car drivers drive with excessively low tire pressure. Of these, 26.5 % were driving with clearly (down to 0.3 bar) and 7.5 % with significantly reduced pressure (0.75 bar and higher). The upshot of excessively low tire pressure is that fuel is wasted.

Tire designation

Definition of terminology

According to EU Directives ECE 30 (for passenger cars, [5]), ECE 54 (for trucks, [6]) and ECE 75 (for motorcycles, [7]) tires must be provided with internationally agreed, standardized tire designations. This applies in particular to the tire sidewall (Figure 18). Tires tested in accordance with ECE (mandatory since October 1998) carry a burned-in circle with a capital "E" or a lower-case "e" and the code of the approving authority, e.g. E4, and followed by a release number (homologation number, see Figure 20).

The lettering, codes and symbols on the sidewall indicate, as well as the tire manufacturer and the type designation, the origin, the production date, the dimension, the load capacity, the maximum permissible speed, the tire design, and the ratio of tire width to tire height (tire cross-section).

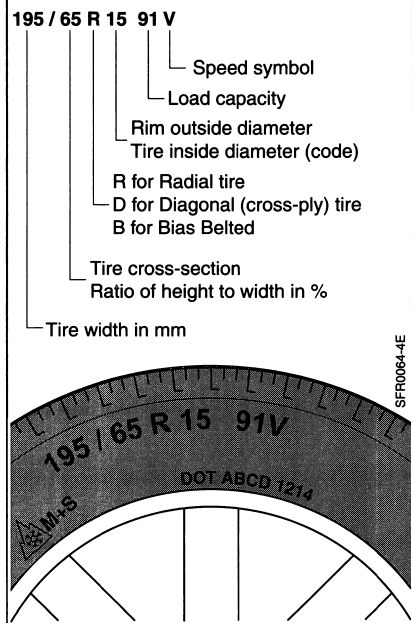
Reading the information is made more difficult by the fact that units of measurement of the metric system standard in Central Europe (mm, bar) and of the British imperial system (1 inch = 25.4 mm) are jointly used.

The service description must feature near the size specification. This consists of the load index (LI, Table 1) and the speed symbol (SSY; also used is SI, speed index, Table 2) and provides information about the maximum load capacity of the tire in question for the top speed corresponding to the speed symbol. This regulation is binding in all EU member states and in Switzerland.

On tires with directional treads (often on winter and summer tires with V-shaped treads) an arrow on the sidewall indicates the prescribed rolling direction. If the tires are already mounted on the rim, they can only be mounted on one side of the vehicle and may not for example be exchanged cross-wise (diagonally) together with the rim.

Example: A 205/55 R 16 91 H size tire has a nominal width of 205 mm, the height of the sidewall is 55 % of the nominal width, which in this case is roughly 112 mm. The diameter of the matching rim to be mounted is 16 inches, i.e.

Figure 18: Tire designation (example)



406 mm. The load capacity corresponds with a load index LI = 91 to the table value 615 kg – the vehicle axle load therefore according to the certificate of registration must not exceed 2 x 615 kg, i.e. 1,230 kg. With the speed index SI = H the vehicle may be driven with this tire at a speed not exceeding 210 km/h, even if the vehicle is designed to reach a higher final speed. Load-capacity reductions must be taken into account from speed index V and upwards.

Production date

There is usually a pressed-in four-digit number in an oval field on at least one of the two sidewalls next to the acronym DOT (US Department of Transportation) and a sequence of letters (code for manufacturing plant). This number denotes the production date. The first two digits indicate the calendar week, the last two

digits the final digits of the production year (Figure 18).

Table 2: Speed index SI (speed symbol)

A1	up to 5 km/h	L	up to 120 km/h
A2	up to 10 km/h	M	up to 130 km/h
A3	up to 15 km/h	N	up to 140 km/h
A4	up to 20 km/h	P	up to 150 km/h
A5	up to 25 km/h	Q	up to 160 km/h
A6	up to 30 km/h	R	up to 170 km/h
A7	up to 35 km/h	S	up to 180 km/h
A8	up to 40 km/h	T	up to 190 km/h
B	up to 50 km/h	U	up to 200 km/h
C	up to 60 km/h	H	up to 210 km/h
D	up to 65 km/h	V	up to 240 km/h
F	up to 80 km/h	W	up to 270 km/h
G	up to 90 km/h	ZR	over 240 km/h
J	up to 100 km/h	Y	up to 300 km/h
K	up to 110 km/h	(Y)	over 300 km/h

Table 1: Load index LI (values up to 3,350 kg, table open in upward direction)

LI	kg	LI	kg	LI	kg	LI	kg	LI	kg	LI	kg
1	46.2	26	95	51	195	76	400	101	825	126	1,700
2	47.5	27	97.5	52	200	77	412	102	850	127	1,750
3	48.7	28	100	53	206	78	425	103	875	128	1,800
4	50	29	103	54	212	79	437	104	900	129	1,850
5	51.5	30	106	55	218	80	450	105	925	130	1,900
6	53	31	109	56	224	81	462	106	950	131	1,950
7	54.5	32	112	57	230	82	475	107	1,000	132	2,000
8	56	33	115	58	236	83	487	108	1,030	133	2,060
9	58	34	118	59	243	84	500	109	1,060	134	2,120
10	60	35	121	60	250	85	515	110	1,090	135	2,180
11	61.5	36	125	61	257	86	530	111	1,120	136	2,240
12	63	37	128	62	265	87	545	112	1,150	137	2,300
13	65	38	132	63	272	88	560	113	1,180	138	2,360
14	67	39	136	64	280	89	580	114	1,215	139	2,430
15	69	40	140	65	290	90	600	115	1,250	140	2,500
16	71	41	145	66	300	91	615	116	1,285	141	2,575
17	73	42	150	67	307	92	630	117	1,320	142	2,650
18	75	43	155	68	315	93	650	118	1,360	143	2,725
19	77.5	44	160	69	325	94	670	119	1,400	144	2,800
20	80	45	165	70	335	95	690	120	1,450	145	2,900
21	82.5	46	170	71	345	96	710	121	1,500	146	3,000
22	85	47	175	72	355	97	730	122	1,550	147	3,075
23	87.5	48	180	73	365	98	750	123	1,600	148	3,150
24	90	49	185	74	375	99	775	124	1,650	149	3,250
25	92.5	50	190	75	387	100	800	125	1,700	150	3,350

Example: 1214 means the 12th week of the year 2014. Prior to the year 2000 the designation of the production date had only three digits.

Special case:

Winter-tire designation

M+S tires

Winter tires must carry the M+S symbol (Mud and Snow) (see Figure 18). EU Regulation No. 661/2009 [8] denotes as an M+S tire a tire whose tread pattern, tread compound or construction is designed first and foremost “to achieve compared with a summer tire better values for winter handling performance and traction on snow” – an extremely woolly definition.

To compare: The previously applicable EU Regulation from 1992 [9] stated that “M+S tires” are such tires “on which the pattern of the tread and the structure are designed in such a way that they guarantee above all in mud and fresh or melting snow better handling performance than normal tires. The pattern of the tread of M+S tires is generally characterized by larger tread grooves and lugs which are separated from each other by larger spaces than is the case on normal tires”.

“M+S” is to date not a protected or precisely defined designation and for this reason may also be used on tires not suitable for winter driving conditions (i.e. also on summer tires). The M+S symbol no longer has any significance with regard to suitability for winter driving conditions.

Snow-flake symbol

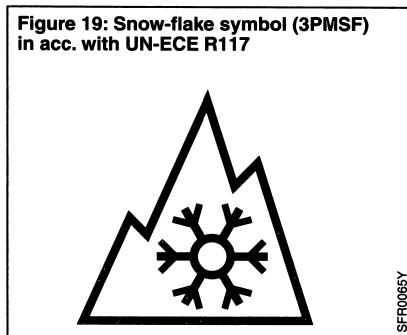
A revision of the M+S designation was called for in response to the widespread traffic chaos in the USA in 1995. A winter tire should satisfy certain criteria with regard to its suitability for winter driving conditions and verify this by way of appropriate tests. This gave rise to the designation 3PMSF (Three Peak Mountain Snow Flake), which today is firmly established in North American legislation.

This designation has also been used voluntarily for a few years in Europe. This is intended to verify to the consumer suitability for winter driving conditions substantiated by tests. The test criteria are defined by the European Union in UN-ECE R 117 [11].

Amendment of StVZO §36 section 4 [12]
With the 52nd Regulation to amend road-traffic legislation of 18 May 2017 the definition of tires for winter weather conditions was adapted and consequently a new winter-tire regulation came into force. By force of this regulation the snow-flake symbol for winter tires is already mandatory for the winter season 2017/2018 in Germany and consequently replaces the M+S marking. In a transition phase lasting until 30 September 2024 tires which only have an M+S marking which continue to be classed as winter tires provided their date of manufacture is before 31 December 2017. M+S tires which are manufactured from 2018 onwards and only have an M+S marking will no longer be classed as winter tires.

For truck and bus tires on the steering axle the introduction of this regulation will be deferred until no later than 1 July 2020. This time will be used to check as part of a study the necessity of the new winter-tire regulation for the steering axle of these vehicle categories. The deferment of the winter-tire regulation does not apply to the permanently driven axles of these vehicles.

Figure 19: Snow-flake symbol (3PMSF) in acc. with UN-ECE R117



Sound identification

Tires featuring this marking comply with Directive ECE 2001/43 [10], which lays down the maximum values for rolling noise. This marking has been mandatory since 1 October 2009, is featured next to the UN/ECE approval mark, and can be recognized by the code letter "s" after the homologation number.

With the introduction of the tire label on 1 November 2012 this designation for new tire types was extended by a capital "S" and by the letters "W" for wet grip and "R" for rolling resistance (Figure 20). Additional code numbers (1 and 2) after the letters S and R denote the limit values which must be adhered to for the prescribed time phases (periods) (regulations, see UN/ECE Regulation No. 117 [11]).

Figure 20: E designation

Example of a homologation number acc. to ECE R 117 for complying with rolling noise.



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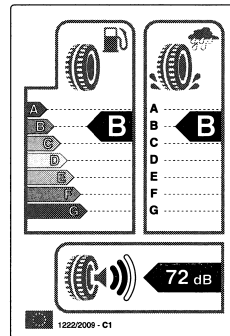
EU tire label

Definition of terminology

Since November 2012 new tires sold in the EU which were produced from July 2012 onwards must be provided with a standardized tire label (7.5 cm × 11 cm) (Figure 21). This label is meant to provide the buyer with quick and unmistakable information on the three tire properties of rolling resistance, wet grip (restricted to wet braking distance) and pass-by noise (not passenger compartment), thereby helping them to make a more informed decision to purchase. The winter properties of winter tires are currently not recorded by the EU tire label.

The tire label records tires in the categories C1 (passenger cars), C2 (light utility vehicles) and C3 (heavy commercial vehicles) as set out in Table 3. Excluded from the regulation are retreaded tires,

Figure 21: EU label



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Table 3: Energy-efficiency classes of the different tire categories

Class C1 tires (passenger cars)		Class C2 tires (light utility vehicles)		Class C3 tires (heavy commercial vehicles)	
c_{RR} in kg/t	Energy-efficiency class	c_{RR} in kg/t	Energy-efficiency class	c_{RR} in kg/t	Energy-efficiency class
$c_{RR} \leq 6.5$	A	$c_{RR} \leq 5.5$	A	$c_{RR} \leq 4.0$	A
$6.6 \leq c_{RR} \leq 7.7$	B	$5.6 \leq c_{RR} \leq 6.7$	B	$4.1 \leq c_{RR} \leq 5.0$	B
$7.8 \leq c_{RR} \leq 9.0$	C	$6.8 \leq c_{RR} \leq 8.0$	C	$5.1 \leq c_{RR} \leq 6.0$	C
not assigned	D	not assigned	D	$6.1 \leq c_{RR} \leq 7.0$	D
$9.1 \leq c_{RR} \leq 10.5$	E	$8.1 \leq c_{RR} \leq 9.2$	E	$7.1 \leq c_{RR} \leq 8.0$	E
$10.6 \leq c_{RR} \leq 12.0$	F	$9.3 \leq c_{RR} \leq 10.5$	F		F
$c_{RR} \geq 12.1$	G	$c_{RR} \geq 10.6$	G		

Example: $c_{RR} = 10.5$ kg/t (label class E) corresponds to $c_{RR} = 0.0105$ or $F_{RR} = 105$ N.

professional off-road tires, racing tires, spikes, compact spare-wheel tires, vintage-car and modern classic-car tires (for vehicles first registered before 1 October 1990), tires for top speeds below 80 km/h, tires with inside diameters of less than 254 mm or more than 635 mm, and motor-cycle tires.

Naturally, the tire label cannot show all the tire criteria (there are up to fifty), but the criteria selected do represent a certain combination of many other linked properties.

Fuel consumption

Letters from A (highest efficiency) through G (lowest efficiency) and the traffic-light colors green, yellow and red in the EU label denote the efficiency of the tire with regard to rolling resistance and thus fuel consumption. The range for rolling resistance from class A through G represents a difference in fuel consumption of up to 7.5% [13]. The potential for savings is even higher, depending on the driving situation. The difference between the individual stages of the tire classes with regard to rolling resistance is clearly defined: for instance 0.11 l/100 km for a vehicle with an average consumption of approx. 6.6 l/100 km. The difference between an A-rated tire and a G-rated tire adds up to approx. 0.5 l/100 km (in each case provided a consistent driving style and identical tire inflation pressure).

Wet grip

Letters A (shortest braking distance) through G (longest braking distance) provide information on the tire's wet grip when braking. The difference in wet grip between a particularly good tire and a poor tire when applying full brakes from 80 km/h to zero produces an 18-meter shorter braking distance.

Tire noise

Pass-by noise is depicted in the EU label by a symbol together with an indication of the dB value. The symbol is based on the binding noise-emission limit values. The more sound waves are shaded black in the EU label, the louder the tire.

Winter tires

Technical characteristics

Passenger-car winter tires exhibit no or only slight differences in structural design from summer tires. They are characterized by good force transmission (traction) on snow and mud, satisfactory grip on ice, good adhesion on wet and dry road surfaces, safe handling, comfortable rolling, and low noise.

The feature that is unique to them is primarily the softer and elastic-when-cold rubber compound with high natural-rubber content. Unlike the summer rubber compound, it does not become brittle and harden ("vitrify") at minus temperatures (which results in reduced adhesion because the interlocking effect can no longer occur).

Added to this are increased tread area (thus a greater positive proportion) and, as a conspicuous external identifying feature, a multidimensional superfine tread (design shapes: zigzag, spheres, honeycombs, and mixed shapes, see Figure 4) in the rubber blocks themselves. These sipes (up to 2,000, depending on tire size) offer additional gripping edges in the snow and increase both traction and braking performance noticeably.

As the tread depth decreases, but also with increasing age (hardening caused by among others the influence of UV and ozone) the winter properties diminish: Traction and cornering-stability potential deteriorate, the braking distance is extended, and the risk of aquaplaning increases. Winter tires identified for example in Austria under a remaining tread depth of 4 mm are always classed as summer tires.

Standard winter tires customary in Europe are offered in speed ratings up to speed index W (maximum speed 270 km/h).

Development aims

The tire is always a compromise product: If a particular property (e.g. rolling resistance) is predominantly developed, this is inevitably at the expense of other properties (e.g. with the upshot of reduced wet grip) and thus of balance. This is only desirable or permissible in special cases (motor sport, special tires, industrial requirements). There is a conflict of aims when on principle contrasting properties are to be simultaneously optimized. Modern rubber compounds with a high silica filler content and optimized ground contact patches raise this balancing act to a higher level, but this does not eliminate the conflict.

The conflicting development aims which can negatively influence each other include grip potential and rolling resistance, dry braking and wet braking, aquaplaning and dry handling, and grip and abrasion.

References for tires

- [1] Source: Goodyear Dunlop, 2012.
- [2] Council Directive 89/459/EEC of 18 July 1989 on the approximation of the laws of the Member States relating to the tread depth of tyres of certain categories of motor vehicles and their trailers.
- [3] Source: Continental AG, 2011. The specified brake differences were determined with a Mercedes C-Class car on 205/55 R 16 V size tyres in over 1,000 brake tests.
- [4] Source: Michelin tire plants, 2010.
- [5] ECE 30: Regulation No. 30 of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of pneumatic tyres for motor vehicles and their trailers.
- [6] ECE 54: Regulation No. 54 – Uniform provisions concerning the approval of pneumatic tyres for commercial vehicles and their trailers.
- [7] ECE 75: Regulation No. 75 – Uniform provisions concerning the approval of pneumatic tyres for motor cycles and mopeds.

Tire tests

Tire manufacturers conduct around fifty objective laboratory tests and subjective road tests on in-house test and racing circuits. The most important tests are:

Handling on a dry road

Criteria are directional stability, steering precision, straight-running stability, noise, and tire comfort.

Properties in the wet

Criteria are handling on a wet course with bends, braking, aquaplaning, longitudinal and lateral, and circular-course driving.

Machine tests

Criteria are top speed, continuous running, and abrasion.

Winter properties

Criteria are snow handling, driving on passes, traction measurement, accelerating performance, and braking.

- [8] Regulation (EC) No. 661/2009 of the European Parliament and of the Council of 13 July 2009 concerning type-approval requirements for the general safety of motor vehicles, their trailers and systems, components and separate technical units intended therefore.

- [9] Council Directive 92/23/EEC of 31 March 1992 relating to tyres for motor vehicles and their trailers and to their fitting installation.

- [10] Directive 2001/43/EC of the European Parliament and of the Council of 27 June 2001 amending Council Directive 92/23/EEC relating to tyres for motor vehicles and their trailers and to their fitting.

- [11] Regulation No. 117 of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of tyres with regard to rolling sound emissions and/or to adhesion on wet surfaces and/or to rolling resistance.

- [12] §36 StVZO: Bereifung, Laufflächen (Tires, treads).

- [13] Source: Tire manufacturers.

Tire-pressure monitoring systems

Application

Tire-Pressure Monitoring Systems (TPMS) are used to monitor the tire pressure on vehicles to prevent tire defects due to insufficient tire pressure, thus reducing the number of accidents resulting from defective tires.

If a vehicle is operated with insufficient tire pressure, this leads to increased flexing energy on the tire sidewalls and thus to increased wear of the tire. When operated at full load or at high speed, the greater flexing energy results in increased thermal load, which can even cause tire bursts. Following a spate of severe fatal accidents in the USA due to tire bursts caused by insufficient inflation pressure, legislation was passed (NHTSA Tread Act) to regulate the nationwide introduction of tire-pressure monitoring systems in the USA in order to warn drivers about low tire inflation pressure at an early stage in the future. Since September 2007, all new cars have been required to be fitted with tire-pressure monitoring systems which detect both tire damage and slow pressure losses through the tire rubber caused by gas diffusion.

The tire inflation pressure, however, is not only an important variable for traffic safety. Ride comfort, tire service life, and fuel consumption are also significantly influenced by the inflation pressure. Inflation pressure reduced by 0.6 bar can increase fuel consumption by up to 4% in urban traffic and shorten the service life of the tire by up to 50%. In the European Union (EU), the decision has been taken to prescribe the fitting of tire-pressure monitoring systems in all new cars as from October 2012 in order to help reduce CO₂ emissions.

Already today the rising proportion of tires with run-flat properties necessitates the deployment of tire-pressure monitoring systems, as the car driver is no longer able to detect a tire with a considerable pressure deficiency ("flat") on the basis of the drivability. In order to prevent the driver from inadvertently exceeding the speed and range limits that apply in this case, run-flat tires may only be used in conjunc-

tion with tire-pressure monitoring systems.

As a general principle, two different types of tire-pressure monitoring system are used: directly measuring and indirectly measuring systems.

Directly measuring systems

In directly measuring systems, a sensor module with a pressure sensor is installed in each tire of the vehicle. This transfers data such as the tire pressure and tire temperature from inside the tire across a coded high-frequency transmission link to a control unit. The control unit evaluates these data in order to detect not only pressure losses in individual tires ("puncture detection") but also slow pressure losses in all the tires ("diffusion detection"). If the tire pressure falls below a specified threshold or if the pressure gradient exceeds a certain value, the driver is warned by a visual or acoustic signal.

The sensor modules are usually integrated into the tire valve. As a rule, they are supplied by a battery. In comparison with other applications, this results in additional requirements with regard to power consumption, media resistance, and sensitivity to acceleration. Micromechanical absolute-pressure sensors are used as sensor elements.

The data measured with the pressure and temperature sensor in the tire are processed in the sensor module, modulated on a HF carrier signal (433 MHz in Europe, 315 MHz in the USA), and emitted via an antenna. This signal is either detected via individual antennas on the wheel arches or in a central receiver (e.g. in the control unit of existing Remote Keyless Entry systems).

Directly measuring systems do not need a reset function if they have a fixed, constant pressure-loss warning threshold. For such a system to work, the vehicle must only have one prescribed inflation pressure, regardless of vehicle load and tire size. As soon as different inflation pressures have to be set in the vehicle, a directly measuring system also needs a reset function to be able to adapt the warning threshold accordingly.

The advantages of directly measuring systems are that they provide precise,

real measurement of the tire pressure and temperature, and their functioning is not dependent on specific tire types, vehicle conditions and road conditions. The disadvantages of direct systems compared with indirect systems are the much higher system costs, the additional logistical costs in the field of maintaining the availability of all the design variants, the follow-up costs for each new rim, and their battery-dependent, limited service life.

Indirectly measuring systems

In indirectly measuring systems, pressure loss in the tires is not determined directly, but rather by means of a derived variable. To achieve this, these systems perform a mathematical-statistical evaluation of the speed differences of all wheels for "puncture detection", and if necessary also an evaluation of a wheel natural-frequency shift for "diffusion detection". In vehicles with antilock braking or driving-dynamics control systems, the wheel speed required for this is determined by sensors that are already present and transferred to the control unit. Speed differences occur when pressure loss reduces the rolling circumference of the corresponding tire, thus increasing its speed relative to the other three wheels. The subtraction, which can be implemented using a low-cost extension of the software algorithms in the antilock braking or driving-dynamics control system, enables detection of high pressure losses on up to three tires. The wheel natural-frequency spectrum of the individual wheels is evaluated to enable a simultaneous pressure loss at all four wheels to be detected. Typically, the maximum wheel natural frequency shifts at a pressure loss of 20% from 40 Hz to approximately 38 Hz.

Indirectly measuring systems must necessarily be calibrated to the nominal pressure. A calibration is initiated by operating the reset button. When the reset function is activated, the system stores the current learning values on the next few kilometers as new reference values, based on the current rolling circumferences and wheel natural-frequency characteristics. The warning capability takes effect after approx. ten minutes of driving time.

The driver is required to activate the reset function to recalibrate the system when one or more tires are changed, the tire positions are changed (e.g. switching the front and rear wheels), the tire pressure is altered (e.g. when the vehicle is fully laden), or work has been carried out on the wheel suspension (e.g. adjustment work, shock-absorber replacement).

Advantages of indirectly measuring systems are their lower system costs and their robustness over the service life of vehicles, since no additional components are required. Because the system is linked to the vehicle and not to the wheels, no further costs are incurred in the field for logistics and spare parts. A disadvantage is the system's dependence on the specific tire, resulting in a wider variation of the detection times and in higher costs for adapting the system's function to the tire dimensions permitted for a particular vehicle. The system's dependence on mileage and road surface also influences the detection times.

Fulfillment of statutory requirements

As things stand, both systems satisfy the statutory requirements in North America and Europe with regard to "puncture detection" and "diffusion detection". Statutory provisions for tire-pressure monitoring systems are also being drafted in China and Korea.

Rotary seal for tire-pressure control

Function

The Turcon PTFE rotary seal (Turcon Roto L, Figure 1) was developed for tire-pressure control purposes in order to seal around the axle of the central tire-pressure system only when required and always when the tire pressure is increased or decreased. In conventional seal concepts the seal remains in permanent contact with the axle that bears the seal. This causes friction, which in turn leads to increased fuel consumption. By avoiding friction in pressureless operation this rotary seal necessarily reduces fuel consumption.

Application

The Turcon Roto L was originally developed primarily for off-road vehicles to adapt the tire inflation pressure to the ground. On asphalt surfaces a high tire inflation pressure is required, whereas on dirt tracks a low tire inflation pressure is advantageous.

The use of this tire-pressure control can also produce benefits for trucks. It facilitates the extension of tire-pressure control systems and produces benefits for trucks with long loading areas or different semi-trailers in which the tire pressure could be optimized for the respective road and load conditions. This would increase traction

and safety and help to deliver significant fuel savings to transport companies.

Performance and passenger cars are a further target market. When sports cars are driving at low speeds the tire pressure should be kept low. At high speeds the handling performance is improved if the tire pressure can be set to high or, depending on the road surface, set differently at each wheel. Furthermore, it can be used as a safety system in the event of a blow-out in order to inflate the system with air until the driver reaches the nearest repair shop.

Design

The Turcon Roto L combines a polytetrafluoroethylene (PTFE) sealing lip with an elastomer seal body and a stable, shape-forming metal ring (Figure 2). The seal's design ensures that during pressure build-up the sealing lip is pressed against the mating sealing surface. As the pressure decreases, the relaxing elastomer area, which acts like a spring, returns the sealing lip to its neutral initial position.

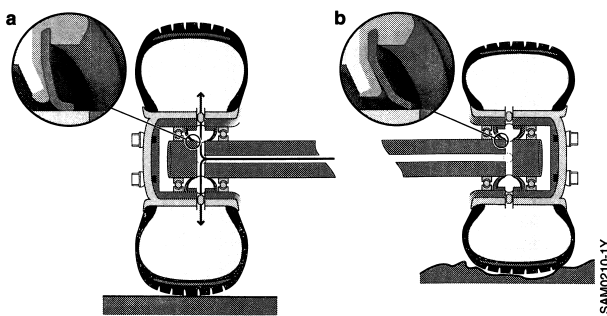
Seal wear

A further important effect on the performance of the Turcon Roto L with regard to the service life of the overall system is the significantly reduced shaft run-in. This is due on the one hand to the use of a special seal material which is filled with a complex mixture of non-abrasive mineral fibers and on the other hand to the low friction when the seal is not activated.

A conventional standard seal used today in a tire-pressure control system already exhibits a shaft run-in of around 10 μm after 168 hours. In comparison, the Turcon Roto L has a shaft run-in of around 4 μm after 780 hours (Figure 3). In other words, the service life of a sealing system with this rotary seal is at least four times longer.

Figure 1: Turcon Roto L sealing system

- a) Setting a high tire inflation pressure,
b) Setting a low tire inflation pressure.



Seal seat

Rotary seals for tire-pressure control need to be well seated in the groove to ensure tightness, even when the system is pressurized. In this way, a relative movement of the body is also prevented when the shaft is subject to axial movement. The groove should have a diameter tolerance of H8 (according to DIN 3760 [1], ISO 6194-1 [2], and ISO 16589-1 [3]). It is important that the seal has not been subjected to load by other components beforehand (for example, by bearings).

The shaft should have a diameter tolerance of h7. The surface should be ground to a perfectly flat finish. A surface with $Ra = 0.2$ to $0.4 \mu\text{m}$ and a maximum Rz value of $1.5 \mu\text{m}$ is recommended for this rotary seal.

Benefits

One benefit of this concept is that it extends the seal life. If the seal in a vehicle is only used for around 10 % of the axle system's service life, it remains operational and fit for use over the system's entire service life.

Even more important, however, is the elimination of friction when there is no pressurization. This means considerable fuel savings for the vehicle operator. Because the seal generates less than half the friction of conventional seal solutions,

the tire pressure can also be reduced or increased during driving. In the case of large commercial-vehicle tires it can take 20 to 30 minutes to adjust the tire pressure. If this can be done en route while the vehicle is driving, the vehicle can be active for up to 30 minutes instead of standing idle. This means a reduction of overall costs for the operator.

References for rotary seal for tire-pressure control

- [1] DIN 3760: Rotary shaft lip type seals; 1996.
- [2] ISO 6194-1: Rotary shaft lip-type seals incorporating elastomeric sealing elements – Part 1: Nominal dimensions and tolerances; 2007.
- [3] ISO 16589-1: Rotary shaft lip-type seals incorporating thermoplastic sealing elements – Part 1: Nominal dimensions and tolerances; 2011.

Figure 2: Turcon Roto L design

- 1 Metal ring,
- 2 Sealing lip.

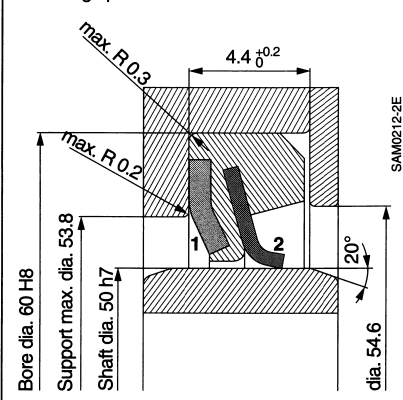
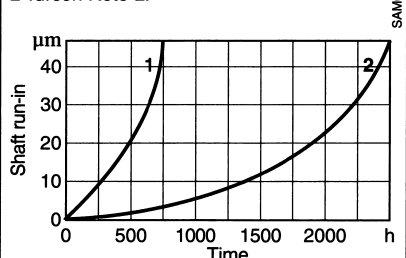


Figure 3: Shaft run-in

- 1 Radial shaft seal,
- 2 Turcon Roto L.



Steering

Purpose of automotive steering systems

The direction of vehicles of any kind is changed with the steering. In single-track vehicles, for example motorcycles, steering is performed by turning the pivoted front fork in the steering head (steering-head steering). In two-track vehicles kingpin steering as described in the following is used.

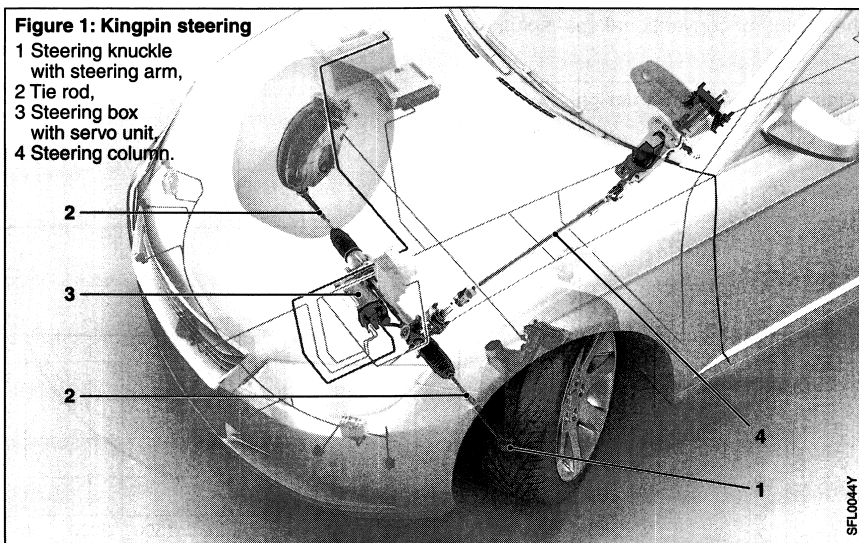
In road vehicles the wheels on the front axle are steered; in some vehicle models the wheels on the rear axle also steer. Rear-axle steering reduces the turning circle at low speeds and increases the driving dynamics at high speeds.

Kingpin steering

In a kingpin-steering system the two wheels are each secured to a pivoted steering knuckle (Figure 1). The rotary movement is introduced via the steering arms rigidly connected to the steering knuckle by the two tie rods actuated by the steering box. The casing of the steering box is permanently connected to the vehi-

cle. The rotary movement is introduced via the steering column, at the upper end of which the steering wheel is located.

Aside from the vehicle weight and the friction coefficient of the wheels on the road surface, the axle kinematics affect the steering behavior and the forces to be applied by the steering. As a result of the toe-in (see Fundamentals of automotive engineering) the two front wheels are not parallel but instead are slightly inclined towards each other in the direction of travel in order to improve straight-running stability. As a result of the camber the wheels are not exactly vertical on the road but instead are slightly inclined inwards. This results in a side force on the wheels acting towards the inside of the vehicle. This force pretensions the bearings in the straight-ahead position and provides a better steering response and greater directional stability. The kingpin inclination influences the steering forces. In the transverse direction the swivel axis of the steering knuckle is inclined inwards, equating to a positive kingpin angle. A small kingpin angle reduces the steering



forces, whereas a large kingpin inclination assists the automatic return to the straight-ahead position.

The automatic return to the straight-ahead position is achieved above all by the caster offset. Inclining the swivel axes in the direction of travel towards the rear moves the piercing point of the swivel axis through the road surface in the direction of travel ahead of the wheel contact point and consequently the wheels are pulled in the direction of travel.

Classification of automotive steering systems

Muscular-energy steering system

The required steering forces are generated exclusively by the muscular energy of the driver. These steering systems are currently used in the smallest vehicle categories.

Power-assisted steering system

The steering forces are generated by the muscular energy of the driver and by an additional auxiliary force hydraulically and increasingly electrically. This steering system is currently the type typically used in passenger cars and commercial vehicles.

Power-steering system

The steering forces are generated exclusively by non-muscular (external) energy (e.g., in machinery).

Friction steering system

The steering forces are generated by forces which act on the tire contact patch. The trailing axles in trucks are an example of this type. The transmissions for the steering and auxiliary forces are effected by mechanical, hydraulic or electrical means or by combinations of the three.

Steering-system requirements

General requirements

To ensure that the road wheels roll cleanly and are thereby not subjected to excessive tire wear, the entire steering kinematics must satisfy the Ackermann condition. This means that the extensions of the wheel axes of the steered wheels intersect at the same point on the extension of the wheel axis of the non-steered wheels (Figure 2).

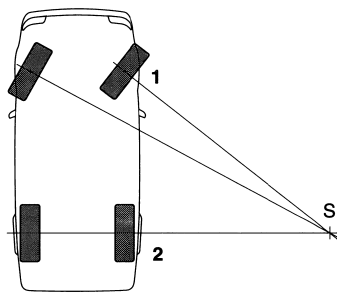
Steering kinematics and axle design must be such that, although the driver receives feedback on the adhesion between wheels and road surface, the steering wheel is not subjected if possible to any forces from the spring motion of the wheels or from motive forces (front-wheel drive). Furthermore, damping properties must be utilized to ensure that no vibration, caused by road irregularities for example, occurs in the steering train.

Jolts from irregularities in the road surface should be damped as much as possible during transmission to the steering wheel. But, in the process, the required haptic feedback from the road to the driver must not be lost.

The steering-angle requirement for turning the steering wheel from lock to lock should, for comfort reasons, be as small as possible when parking and driving at low speed. However, the direct

Figure 2: Ackermann condition

- 1 Front axle,
- 2 Rear axle.
- S Point of intersection of extension of wheel axes.



steering ratio should not make the vehicle susceptible to instability at medium and high speeds.

Statutory requirements

Displacement and time synchronization when the steering system is operated is required by law. A rigid, zero-play configuration of all the components in the steering train helps to achieve this and causes the slightest steering-wheel movements by the driver to be translated into changes of direction at the steered wheel. This property is particularly important in the range of the straight-ahead position. This makes possible safe, precise, and fatigue-free vehicle guidance.

A further statutory requirement is the tendency of the steering to return automatically to the straight-ahead position. This is achieved by appropriate axle kinematics and a low-friction design of the components in the steering train.

The statutory requirements imposed on steering systems in motor vehicles are described in the international regulation ECE-R79 [1]. These requirements include, as well as the basic functional requirements, the maximum permissible control forces for an intact steering system and for a faulty steering system. These requirements govern, above all, the behavior of the vehicle and the steering system when driving into and out of a circle. Owing to different requirements, vehicles are divided in ECE-R79 into different categories, to which different stipulations then apply (Table 1).

For vehicles of all categories: After the steering wheel is released when the vehicle is driven on a circular course at half lock and at a speed of 10 km/h or faster, the driven radius of the vehicle must become larger or at least remain the same.

For M1 category vehicles (passenger cars with up to eight seats in addition

Table 1: Categorization of vehicles in ECE-R79

Vehicle category	
M1	Vehicles for the carriage of passengers comprising no more than 8 seats in addition to the driver's seat
M2	Vehicles for the carriage of passengers comprising more than 8 seats in addition to the driver's seat and having a maximum mass not exceeding 5 tons
M3	Vehicles for the carriage of passengers comprising more than 8 seats in addition to the driver's seat and having a maximum mass exceeding 5 tons
N1	Vehicles for the carriage of goods and having a maximum mass not exceeding 3.5 tons
N2	Vehicles for the carriage of goods and having a maximum mass exceeding 3.5 tons but not exceeding 12 tons
N3	Vehicles for the carriage of goods and having a maximum mass exceeding 12 tons

Table 2: Regulations for steering operating force when driving into a circle at a speed of 10 km/h

Vehicle category	Intact system			Faulty system		
	Maximum operating force in daN ¹	Time in s	Turning circle radius in m	Maximum operating force in daN ¹	Time in s	Turning circle radius in m
M1	15	4	12	30	4	20
M2	15	4	12	30	4	20
M3	20	4	12 ²	45	6	20
N1	20	4	12	30	4	20
N2	25	4	12	40	4	20
N3	20	4	12 ²	45 ³	6	20

¹ 1 daN = 10 N.

² Or steering lock in case this value is not reached.

³ 50 daN for non-articulated vehicles, with two or more steered axles, excluding friction-steered axles.

to the driver's seat): When the vehicle is driven tangentially out of a circle with a radius of 50 m at a speed of 50 km/h, no unusual vibrations may occur in the steering system. In vehicles of the other categories, this behavior must be substantiated at a speed of 40 km/h or, if this value is not reached, at top speed.

This behavior is also prescribed in the event of a fault in vehicles with power-assisted steering systems. For M1 category vehicles, it must be possible in the event of a steering-servo failure to drive at a speed of 10 km/h within four seconds into a circle with a radius of 20 m. The control force at the steering wheel must not exceed 30 daN in the process (Table 2). The same applies to the behavior with an intact system.

Types of steering box

The specified steering-system requirements have given rise above all to two fundamental types of steering box. Both types can be utilized in pure muscular-energy steering systems or, in combination with appropriate servo systems, as power-assisted steering systems.

Rack-and-pinion steering

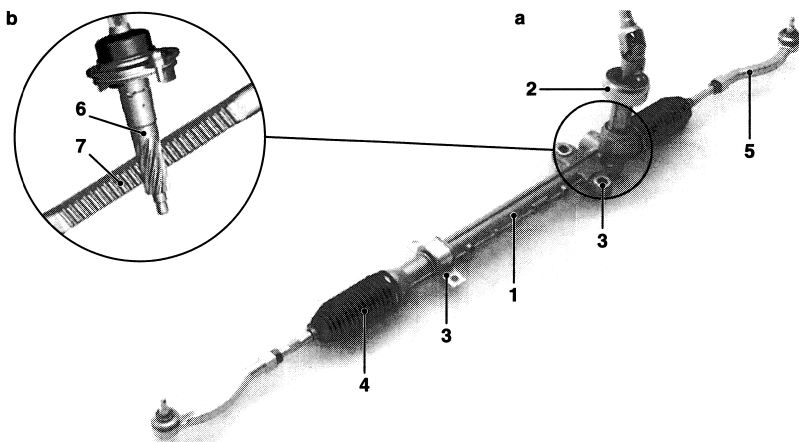
Rack-and-pinion steering is by far the most commonly used type and consists essentially of steering pinion and rack (Figure 3). The steering pinion is driven by the steering wheel and the steering column (Figure 1). The steering ratio is defined by the ratio of pinion revolutions (steering-wheel revolutions) to rack travel.

As an alternative to a constant reduction ratio on the rack, suitable toothing of the rack allows the ratio to be varied as a function of travel. In this way, the straight-running stability of the vehicle can be improved by a suitably indirect

Figure 3: Rack-and-pinion steering

a) Design,
b) Detail.

- 1 Steering-gear box with rack,
- 2 Steering spindle with pinion,
- 3 Steering mounting,
- 4 Gaiter,
- 5 Tie rods with outer links,
- 6 Steering pinion,
- 7 Rack with toothing.



ratio around the center of the steering. At the same time, it is possible with a direct ratio arrangement in the range of medium and large steering angles (e.g., when parking) to reduce the necessary steering-angle requirement when turning from lock to lock.

Recirculating-ball steering

The forces generated between steering worm and steering nut are transmitted via a low-friction row of recirculating balls (Figure 4). The steering nut acts on the steering output shaft via gear teeth. A variable ratio is also possible with this type of steering box.

The increasing performance of rack-and-pinion steering has meant that recirculating-ball steering is practically no longer used in passenger cars.

Power-assisted steering systems for passenger cars

The upshot of the increasing size and weight of vehicles and the heightened comfort and safety requirements involved is that in the last few years power-assisted steering has gained acceptance in all vehicle categories down to the compact-car segment. These steering systems are, but for a few exceptions, already installed as standard. The steering forces exerted by the drive are boosted by a hydraulic or an electric servo system. This servo system must be such that the driver receives good feedback on the adhesion conditions between the tires and the road surface at all times, and yet negative influences caused by road-surface jolts are effectively damped.

Hydraulic power-assisted steering

Combining the mechanical type of steering box with a hydraulic servo system produces rack-and-pinion power steering (Figure 6) and ball-and-nut power steering.

Control valve

The control valve provides the working cylinder with an oil pressure that corresponds to the rotary force of the steering wheel (Figure 5). For this purpose, a torsion-bar spring is used to convert the operating torque into proportional actuator travel inside the valve. The actuator travel changes opening cross-sections inside the valve and thereby control the oil and pressure ratios in the working cylinder.

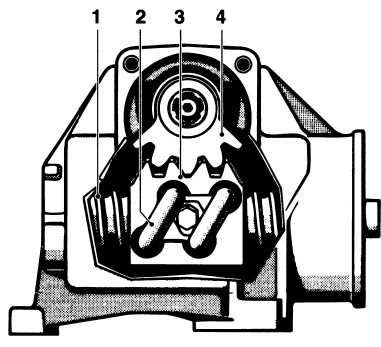
Control valves are built according to the "open center" principle, i.e., when the control valve is not actuated, the oil delivered by the pump flows back to the oil reservoir at zero pressure.

Power-assisted steering systems with modulation capability

Increasing demands regarding user-friendliness and safety have resulted in the introduction of power-assisted steering systems with modulation capability. An example of this is the rack-and-pinion power-steering system operating in response to driving speed. An ECU evaluates the driving speed and controls,

Figure 4: Recirculating-ball steering

- 1 Steering worm,
- 2 Recirculating balls,
- 3 Steering nut,
- 4 Steering output shaft with toothed segment.



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via an electrohydraulic control valve, the hydraulic feedback in the system and consequently the operating force at the steering wheel.

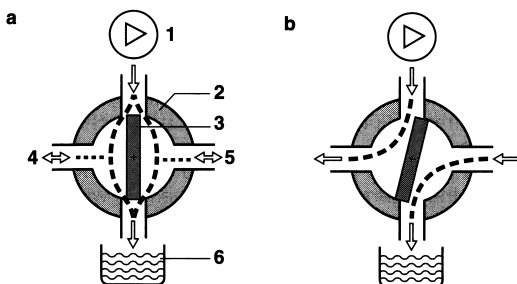
This produces in the parking range and when driving slowly low operating forces, which then increase as the speed increases. In this way, precise and accurate steering is possible at high speeds.

Working cylinder

The double-action working cylinder integrated in the steering box converts the applied oil pressure into an assisting force which acts on the rack and intensifies the steering force exerted by the driver. As the working cylinder has to have extremely low friction, particularly high demands are made on the piston and rod seals.

Figure 5: Functioning principle of control valve of hydraulic power-assisted steering

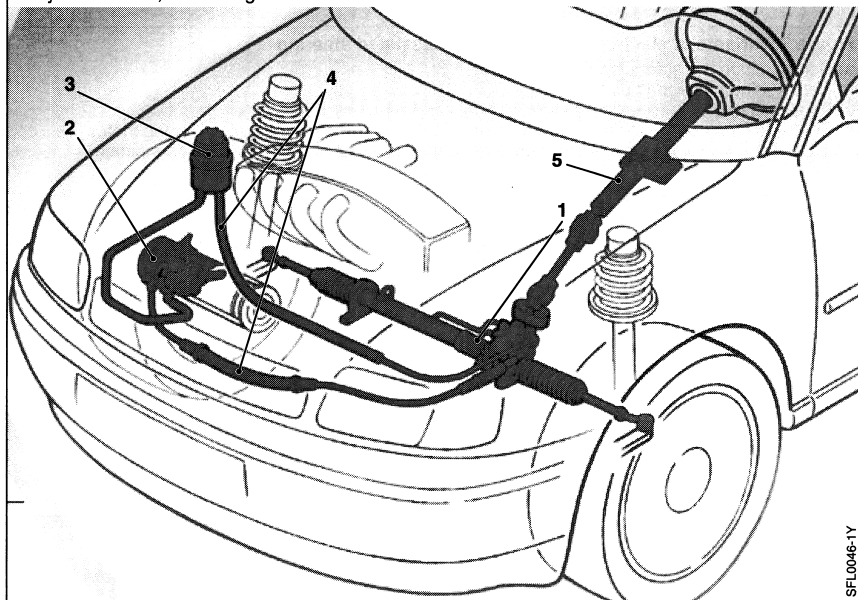
- a) Control valve in neutral position,
b) Control valve in working position.
1 Power-steering pump,
2 Control bushing,
3 Rotary slide,
4 Left working-cylinder chamber,
5 Right working-cylinder chamber,
6 Oil reservoir.



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Figure 6: Rack-and-pinion power steering

- 1 Hydraulic steering box, 2 Power-steering pump, 3 Oil reservoir,
4 Hydraulic lines, 5 Steering column.



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Energy source

The energy source consists of a vane-type supply pump (generally driven by the internal-combustion engine) (power-steering pump) with an integrated oil-flow regulator, an oil reservoir, and connecting hoses and pipes. The pump must be dimensioned so that it makes available the required oil pressure and the required oil quantity for parking maneuvers, even when the engine is just idling.

To provide protection against overload, steering systems contain a pressure-relief valve which is usually integrated in the pump. The pump and the components in the hydraulic working circuit must be designed so that no offending noise is generated and the operating temperature of the hydraulic fluid does not rise to an excessive level.

Alternatively, the pump for the energy supply of the steering can also be driven by an electric motor. The pump is usually designed as a gear or roller-cell pump. The pump can be mounted in a variety of locations due to the omission of the belt drive for the internal-combustion engine. Thanks to an electronic control module which evaluates signals like the vehicle or steering speed, the pump's rotational

speed can be adapted to the steering's current energy demand and to the driving situation, and consequently greater energy efficiency can be achieved.

Electric power-assisted steering

Hydraulic power-assisted steering has been replaced in the passenger-car sector in the last few years almost exclusively by electromechanical power-assisted steering (EPS, Electric Power Steering). Introduced in micro and compact cars in around 1990 and then in the compact class in around 2000, ever more powerful electric ECUs and motors enable such systems to be used even in the upper passenger-car and SUV sectors. Electromechanical power-assisted steering is characterized by an electric drive unit consisting of an ECU and an electric motor powered by the vehicle electrical system which delivers the required servo force. The system consists of the following components (Figure 7):

- Steering column that connects the steering pinion with the steering wheel inside the vehicle.
- Steering pinion that converts the rotating steering movement into the linear movement of the rack.

Figure 7: Variants of electromechanical power-assisted steering

a) Servo unit on the steering column,

b) Servo unit on a second pinion,

c) Paraxial servo unit.

1 Toothing for steering wheel, 2 Upper steering column, 3 Torque sensor, 4 Electric motor,

5 ECU, 6 Helical-gear drive, 7 Intermediate steering shaft, 8 Steering-spindle connection,

9 Mechanical rack-and-pinion steering,

10 Steering box,

11 Gaiter,

12 Tie rod,

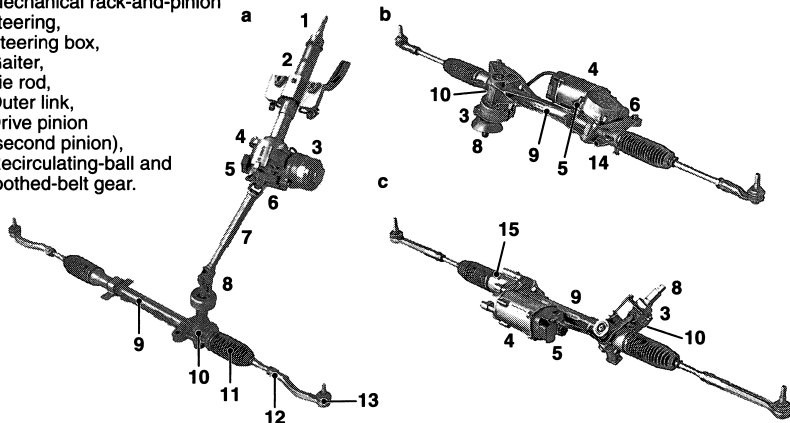
13 Outer link,

14 Drive pinion

(second pinion),

15 Recirculating-ball and

toothed-belt gear.



- Rack connected to the wheels via the tie rods and links.
- Sensors to record the information required to calculate the necessary supporting steering torque.
- Servo unit, consisting of an electric ECU with microprocessor, electric motor and reduction gear which generates the assisting steering torque and links into the steering train.

Variants

The mechanical coupling of the motor to the steering box can be set up as a steering-column, steering-pinion or rack drive.

Servo unit on the steering column

The servo unit is integrated along with its electronics in the steering column (Figure 7a). It is connected via the intermediate steering shaft with universal joints to the mechanical rack-and-pinion steering. The torque generated by the electric motor is converted via the helical-gear drive (Figure 8) into an assisting torque and transmitted to the steering column. The sensor technology and the torsion-bar spring are located next to the helical-gear drive.

This system is used in vehicles with low steering forces (micro and compact cars).

Servo unit on a second pinion

The servo unit is mounted on a second pinion (Figure 7b), meaning that the sensor unit and drive unit can be separate. The helical-gear drive (Figure 8) converts the torque provided by the electric motor into the servo assisting torque and transmits it to the rack. The drive-pinion ratio is not dependent on the steering ratio; this makes possible a power-optimized design with 10 % to 15 % more system power.

This system is used in mid-size vehicles.

Paraxial servo unit

To convert the rotary movement of the steering wheel into a linear motion of the rack, this system uses a gear concept consisting of toothed belt and recirculating-ball gear. A system with ball recirculation is used in the ball-and-screw spindle drive. The ball chain is returned through a channel integrated in the recirculating-ball

nut. The slip-free toothed belt is able to transmit the torque reliably.

This system is used in vehicles with high steering forces (sports cars, upper mid-size, off-road vehicles, light commercial vehicles).

Servomotor

Brushed or brushless DC motors are used as servomotors. Depending on the required performance capability of the steering the design of the reduction gear, the torque generated by these motors is 3 to over 10 Nm.

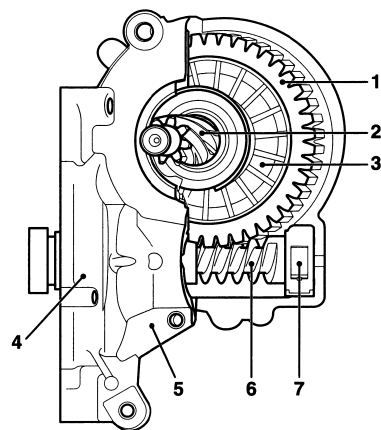
Activation of the motor and hence the power consumption itself are demand-oriented; in other words, only when assisting power is actually required is the vehicle electrical system subjected to load extending beyond the stand-by power input.

Reduction gear

Primarily two principles are used in the reduction gear.

Figure 8: Helical-gear drive

- 1 Helical gear,
- 2 Drive pinion,
- 3 Overload safety,
- 4 Locating bearing,
- 5 Housing,
- 6 Worm,
- 7 Spring-damper element.



Helical-gear drive

When a helical-gear drive is used (Figure 8), the motor drives a worm which engages a helical gear and thereby acts on the steering output shaft, the steering pinion or a separate drive pinion (Figure 8). It is important to ensure in the design of the helical-gear drive that there is no tendency to self-lock when turning back as a result of the rack force.

Recirculating-ball gear

When a recirculating-ball gear is used (Figure 9), the motor drives, either directly or additionally reduced by an upstream belt-transmission stage, a ball-and-screw spindle drive, which converts the rotary movement into linear motion of the rack.

Torque sensor

To deliver the assisting force in line with demand, the driver manual torque is measured by the torque sensor integrated in the steering system, the required servo force is calculated in the electric ECU, and the motor is activated accordingly. The assisting force is delivered by the motor on the basis of torque control (see Torque sensor).

Functions

Further functions are executed in the ECU software to optimize the steering behavior. The ECU evaluates further signals from the vehicle or the steering (e.g., driving speed, steering angle, steering torque, and steering speed) for this purpose. Thus, for example, the re-setting movement of the steering wheel is optimized as a function of the steering angle and the driving speed by an additional motor torque. Likewise, a damping function is implemented with the information on driving speed and steering speed; this function supports the straight-ahead stability of the steering and hence of the vehicle at high driving speeds.

By networking the steering ECU with other ECUs in the vehicle network, it is possible to realize assistance functions with the electromechanical power-assisted steering to enhance comfort and safety.

Demand-oriented control of the electric motor leads to considerable fuel savings of 0.3 l/100 km on average compared to hydraulic power steering with a pump driven by the vehicle engine. In urban driving, the fuel savings increase to 0.7 l/100 km.

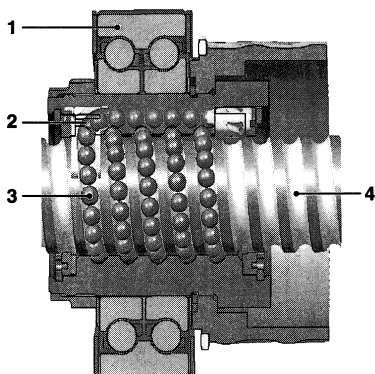
In the event of a failure of the energy supply or steering assistance, the driver can continue to steer mechanically, but with higher manual steering torque.

Superimposed steering

With a superimposed-steering system, an additional steering angle can be added to or subtracted from the steering-wheel angle set by the driver at the steering wheel. This system is usually combined with a parameterizable hydraulic or an electric power-assisted steering system. Superimposed does not facilitate autonomous driving, but it does provide for a steering characteristic optimally adapted to the driving situation, and thus maximum comfort and directional stability. When networked with driving-dynamics control systems, it can further increase safety in critical driving situations by means of driver-independent steering adjustments. Such steering systems are already in series production, known as Active Steering at BMW and Dynamic Steering at Audi.

Figure 9: Recirculating-ball gear

- 1 Ball bearing,
- 2 Return channel,
- 3 Ball chain,
- 4 Rack.



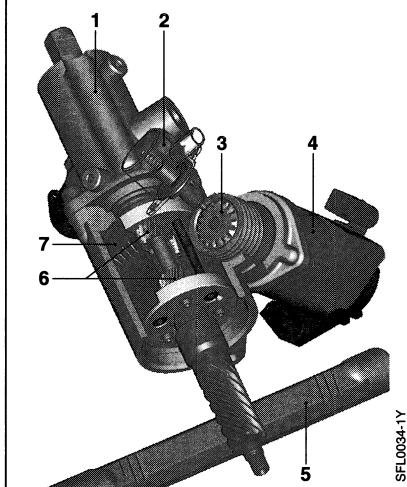
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Technical solution

The angle superimposition independent of the driver's steering angle is currently effected by two different technical solutions.

Figure 10: Planetary-gear set of superimposed steering

- 1 Valve,
- 2 Electromagnetic lock,
- 3 Worm,
- 4 Electric motor,
- 5 Rack,
- 6 Planetary gear,
- 7 Worm gear.

**Planetary-gear set**

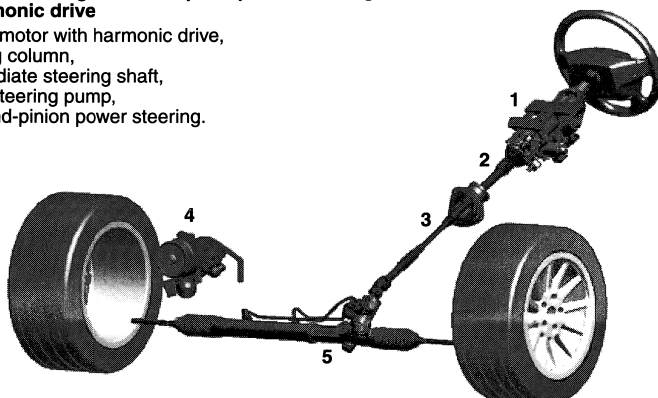
A twin planetary-gear set with different gear ratios of the gear stages is integrated into a common planetary-gear carrier in the steering train (Figure 10). This means there is always a mechanical link between the steering wheel and the steered wheels. The different gear ratios of the gear stages mean that turning the planetary-gear carrier sets the additional steering angle. The angle setting is made by a electric motor that drives the worm gear of the planetary-gear carrier.

Harmonic drive

The steering-angle superimposition unit consists in this case of a harmonic drive and an electric motor with a hollow shaft (Figure 11). The highly compact design enables it to be integrated into the steering column without compromising the requirements with regard to installation space and crash behavior. The shaft at the steering-wheel end (input shaft) is positively connected (Figure 12) to a flex spline. The rotary movement of the steering wheel is transmitted via the toothing of the flex spline (teeth in the region of the flexible ball bearing point outwards) to the toothing of the internal gear (circular spline, teeth point inwards in the direction of the axle) for the output shaft. An elliptical inner rotor (shaft generator) located in the flex spline, which is driven by the electric motor, generates the superimposed steering angle via the different number

Figure 11: Steering train of superimposed steering with harmonic drive

- 1 Electric motor with harmonic drive,
- 2 Steering column,
- 3 Intermediate steering shaft,
- 4 Power-steering pump,
- 5 Rack-and-pinion power steering.



of teeth between the flex spline and the circular spline. The races of the flexible ball bearing follow the shaft motion of the elliptical inner rotor and transmit this shaft motion to the flex spline. Here, too, there is always a mechanical link between the steering wheel and the steered wheels via the toothing of the harmonic drive.

In the passive state, the electric motor is blocked by an electromechanical lock, thereby ensuring direct mechanical through-drive for the steering movement.

Activation concept

The superimposed-steering ECU checks the plausibility of the required sensor information and evaluates this information. It calculates the setpoint angle for the electric motor and generates via an integrated driver stage the pulse width-modulated signals for activating the electric motor. This is implemented as a brushless DC motor with integrated rotor position sensor. The maximum motor current is 40 A at a vehicle system voltage of 12 V. The rotor position sensor enables the control unit to control the electronic commutation and thus the direction of rotation of the motor. It also calculates and checks the total set

additional steering angle using a summation algorithm in the control unit software.

The effective steering angle, the sum total of the steering-wheel angle and the superimposed angle of the electric motor, is calculated by the ECU, and made available on the vehicle communication bus to the partner ECUs.

Setpoint value

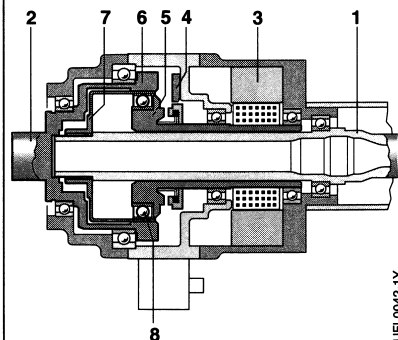
The setpoint value for the effective steering angle formed in the superimposed-steering ECU is made up of the partial setpoint value for steering comfort and the partial setpoint value for vehicle stabilization. The signals required to calculate these variables are read in by the control unit via the CAN bus.

The partial setpoint value for the steering comfort is implemented as a speed-dependent, variable steering ratio. The value is calculated from the input variables of steering angle and driving speed. When the vehicle is stationary and at low driving speeds, an angle is added to the steering angle set by the driver. This makes the steering ratio more direct. The driver can turn the wheels fully with less than one complete steering wheel revolution. This steering-angle addition is continuously reduced as driving speed increases. From speeds of roughly 80 – 90 km/h, a proportion is subtracted from the driver's steering angle and steering becomes more indirect. This ensures straight-running vehicle stability at high speeds and at the same time prevents the driver from losing control of the vehicle due to excessively fast steering movements.

For the calculation of the partial setpoint value for vehicle stabilization – in addition to the steering angle and the driving speed – the vehicle movement is measured using the sensors for the vehicle yaw rate and lateral acceleration. The superimposed steering uses the driving-dynamics control sensors for this purpose. In the same way as the driving-dynamics control, a calculation model that runs in the ECU software calculates the reference vehicle movement. In the event of a deviation of the actual vehicle movement from the reference movement, the steering is activated to stabilize the vehicle. The two systems continuously

Figure 12: Actuator of superimposed steering with harmonic drive

- 1 Input shaft,
- 2 Output shaft,
- 3 Electric motor,
- 4 Rotor-position sensor,
- 5 Elliptical inner rotor (shaft generator),
- 6 Internal gear (circular spline),
- 7 Flex spline,
- 8 Flexible ball bearing.



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exchange information so that the controllers of the driving-dynamics control and the superimposed steering cooperate to optimal effect.

Safety concept

All of the internal and external signals that are used are continuously monitored in the control unit and their plausibility checked. If a sensor signal is no longer plausible, the additional steering function on which it is based is first disabled. For example, if the yaw-rate sensor that measures the rotation of the vehicle around its vertical axis (yaw rate) fails, the yaw-rate control of the superimposed-steering system is disabled. The variable steering ratio remains active.

If safe activation of the electric motor is no longer possible due to a fault, the system is completely shut down and direct steering-wheel through-drive is ensured by self-locking of the gear stage and by an electromechanical lock. This fallback level is always automatically active when the internal-combustion engine is not running or even if there is no electrical supply voltage.

Power-assisted steering systems for commercial vehicles

Power-assisted steering with all-hydraulic transmission

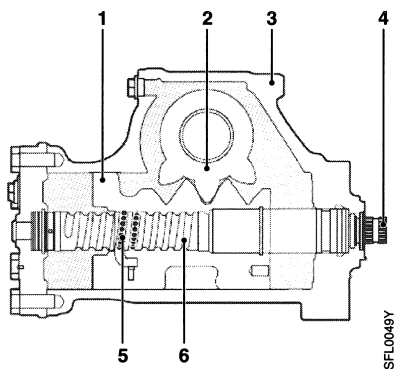
Hydrostatic steering systems are hydraulic power-assisted steering systems. The steering force of the driver is hydraulically boosted and transmitted exclusively by hydraulic means to the steered wheels. Because there is no mechanical connection, the maximum permissible speed is limited by national regulations. In Germany, this is 25 km/h. Depending on the system configuration and emergency steering properties, approval up to a speed of 62 km/h is possible. Use of these systems is therefore confined to machinery and special vehicles.

Single-circuit power-assisted steering system for commercial vehicles

Commercial vehicles are usually equipped with ball-and-nut power steering (Figure 13). The control valve is integrated into the steering box and together with the steering worm forms a single unit. The rotary movement of the steering wheel is transmitted via an endless ball chain to

Figure 13: Ball-and-nut power steering

- 1 Piston,
- 2 Steering-sector shaft,
- 3 Housing,
- 4 Steering-spindle connection,
- 5 Ball nut with ball chain,
- 6 Steering worm.



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the ball-and-nut. Short toothing on the ball-and-nut meshes with the toothing on the segment shaft. The generated rotary movement of the segment shaft is transmitted via a steering arm to the steering linkage and the steered wheels.

The servo force is applied, as with rack-and-pinion power steering, by a rotary-slide valve. The working cylinder is formed by a sealing surface between the ball-and-nut housing and the steering box. Because no additional lines are required outside the housing, a robust and compact steering box with high power output is created.

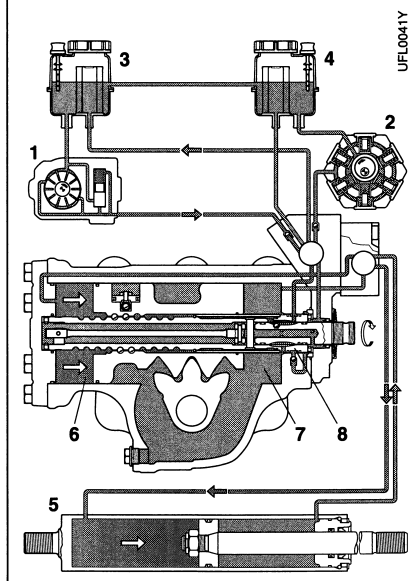
Dual-circuit power-assisted steering system for heavy-duty commercial vehicles

Dual-circuit steering systems (Figure 14) are required when the operating forces needed at the steering wheel exceed the statutory requirements in ECE-R79 [1] if the power-assistance system fails. These steering systems feature hydraulic redundancy. Both steering circuits in these systems are functionally tested by means of flow indicators and a fault is indicated to the driver. The pumps for supplying the independent steering circuits must be driven in various ways (e.g., engine-dependent, driving speed-dependent, or electrically). If a circuit fails, caused for example by a fault in the steering system or engine failure, the vehicle can be steered with the still operational redundant circuit in accordance with the statutory requirements.

Dual-circuit systems take the form of ball-and-nut power steering with an integrated second steering valve. This second valve controls an additionally externally installed working cylinder and thereby provides the redundancy in relation to the existing servo system in the ball-and-nut steering.

Figure 14: Dual-circuit power-assisted steering

- 1 Steering pump 1,
- 2 Steering pump 2,
- 3 Oil reservoir 1,
- 4 Oil reservoir 2,
- 5 Working cylinder,
- 6 Left cylinder chamber,
- 7 Right cylinder chamber,
- 8 Dual-circuit steering valve.



References

- [1] ECE-R79: Uniform provisions concerning the approval of vehicles with regard to steering equipment.

Brake systems

Definitions and principles

(based on ISO 611 [1] and DIN 70024 [2]).

Brake equipment

All the vehicle brake systems whose functions are to reduce vehicle speed or bring the vehicle to a halt, or to hold the vehicle stationary if already halted.

Brake systems

Service brake system

All the elements, the action of which may be regulated, allowing the driver to reduce, directly or indirectly, the speed of a vehicle during normal driving or to bring the vehicle to a halt.

Secondary brake system

All the elements, the action of which may be regulated, allowing the driver to reduce, directly or indirectly, the speed of a vehicle or to bring the vehicle to a halt in case of failure of the service brake system.

Parking brake system

All the elements allowing the vehicle to be held stationary mechanically even on an inclined surface, and particularly in the absence of the driver.

Continuous-operation brake system

System of components which allows the driver to reduce the vehicle's speed or descend a long downhill gradient at a virtually constant speed with practically no wear to the friction brakes. A continuous-operation brake system may incorporate one or more retarders.

Automatic brake system

All the elements which automatically brake the trailer as a result of intended or accidental separation from the tractor vehicle.

Electronic brake system (ELB, EHB)

Brake system controlled by an electrical signal generated and processed by the control transmission system. An electrical output signal controls components which generate the application force.

Component parts

Energy-supplying device

Parts of a brake system which supply, regulate and, if necessary, condition the energy required for braking. It terminates at the point where the transmission device starts, where the various circuits of the brake systems, including the circuits of accessories if fitted, are isolated either from the energy supplying device or from each other.

The energy source is that part of an energy-supplying device which generates the energy. It may be located remotely from the vehicle (e.g. in the case of a compressed-air braking system for a trailer) or may be the muscular force of the driver.

Control device

Parts of a brake system which initiate the operation and control the effect of this brake system. The control signal can be conveyed within the control device by, for example, mechanical, pneumatic, hydraulic or electrical means, including the use of auxiliary energy or non-muscular force.

The control device is defined as starting at the component to which the control force is directly applied. It can be operated:

- By direct application of force by the driver by hand or foot,
- by indirect action of the driver or without any action (only in the case of trailers),
- by varying the pressure in a connecting line, or the electric current in a cable, between the tractor vehicle and the trailer at the time when one of the brake systems on the tractor vehicle is operated, or if it fails,
- by the inertia of the vehicle or by its weight or of one of its main component parts.

The control device is defined as ending at the point at which the braking energy is distributed, or where part of the energy is diverted to control braking energy.

Transmission device

Parts of a brake system which transmit the energy distributed by the control device. It starts either at the point where the control device terminates or at the point where the energy supplying device terminates. It terminates at those parts of the brake system in which the forces opposing the vehicle's movement, or its tendency towards movement, are generated. It can, for example, be mechanical, hydraulic, pneumatic (pressure above or below atmospheric), electric, or combined (for example hydro-mechanical, hydropneumatic).

Brake

Parts of a brake system in which the forces opposing the vehicle's movement, or its tendency towards movement, are developed, such as friction brakes (disk or drum) or retarders (hydrodynamic or electrodynamic retarders, exhaust brakes).

Auxiliary device of the tractor vehicle for a trailer

Parts of a brake system on a tractor vehicle which are intended to supply energy to, and control, the brake systems on the trailer. It comprises the components between the energy supplying device of the tractor vehicle and the supply-line coupling head (inclusive), and between the transmission device(s) of the tractor vehicle and the control-line coupling head (inclusive).

Brake-system types relating to the energy supplying device*Muscular-energy brake system*

Brake system in which the energy necessary to produce the braking force is supplied solely by the physical effort of the driver.

Energy-assisted brake system

Brake system in which the energy necessary to produce the braking force is supplied by the physical effort of the driver and one or more energy supplying devices.

Non-muscular-energy brake system

Brake system in which the energy necessary to produce the braking force is supplied by one or more energy-supplying devices excluding the physical effort of the driver. This is used only to control the system.

Note: A brake system in which the driver can increase the braking force, in a state of totally failed energy, by muscular effort acting on the system, is not included in the above definition.

Inertia brake system

Brake system in which the energy necessary to produce the braking force arises from the approach of the trailer to its tractor vehicle.

Gravity brake system

Brake system in which the energy necessary to produce the braking force is supplied by the lowering of a component part of the trailer (e.g. trailer drawbar) due to gravity.

Definitions of brake systems relating to the arrangement of the transmission device*Single-circuit brake system*

Brake system having a transmission device embodying a single circuit. The transmission device comprises a single circuit if, in the event of a failure in the transmission device, no energy for the production of the application force can be transmitted by this transmission device.

Multi-circuit brake system

Brake system having a transmission device embodying several circuits. The transmission device comprises several circuits if, in the event of a failure in the transmission device, energy for the production of the application force can still be transmitted, wholly or partly, by this transmission device.

Definitions of brake systems relating to vehicle combinations

Single-line brake system

Assembly in which the brake systems of the individual vehicles act in such a way that the single line is used both for the energy supply to, and for the control of, the brake system of the trailer.

Dual- or multi-line brake systems

Assembly in which the brake systems of the individual vehicles act in such a way that several lines are used separately and simultaneously for the energy supply to, and for the control of, the brake system of the trailer.

Continuous brake system

Combination of brake systems for vehicles forming a road train. Characteristics:

- From the driving seat, the driver can operate a directly operated control device on the tractor vehicle and an indirectly operated control device on the trailer by a single operation and with a variable degree of force.
- The energy used for the braking of each of the vehicles forming the combination is supplied by the same energy source (which may be the muscular effort of the driver).
- Simultaneous or suitably phased braking of the individual units of a road train.

Semi-continuous brake system

Combination of brake systems for vehicles forming a road train. Characteristics:

- The driver, from his driving seat, can gradually operate a directly operated control device on the tractor vehicle and an indirectly operated control device on the trailer by a single operation.
- The energy used for the braking of each of the vehicles forming the road train is supplied by at least two different energy sources (one of which may be the muscular effort of the driver).
- Simultaneous or suitably phased braking of the individual units of a road train.

Non-continuous brake system

Combinations of the brake systems of the vehicles forming a road train which is neither continuous nor semi-continuous.

Brake-system control lines

Wiring and conductors: These are employed to conduct electrical energy.

Tubular lines: Rigid, semi-rigid or flexible tubes or pipes used to transfer hydraulic or pneumatic energy.

Lines connecting the brake equipment of vehicles in a road train

Supply line: A supply line is a special feed line transmitting energy from the tractor vehicle to the energy accumulator of the trailer.

Brake line: A control line is a special control line by which the energy essential for control is transmitted from the tractor vehicle to the trailer.

Common brake and supply line:

Line serving equally as brake line and as supply line (single-line brake system).

Secondary-brake line: Special actuating line transmitting the energy from the tractor vehicle to the trailer essential for secondary braking of the trailer.

Braking mechanics

Mechanical phenomena occurring between the start of actuation of the control device and the end of the braking action.

Gradual braking

Braking which, within the normal range of operation of the control device, permits the driver, at any moment, to increase or reduce, to a sufficiently fine degree, the braking force by operating the control device. When an increase in braking force is obtained by the increased action of the control device, an inverse action must lead to a reduction in that force.

Brake-system hysteresis: Difference in control forces between application and release at the same braking torque.

Brake hysteresis: Difference in application force between application and release at the same braking torque.

Forces and torques

Control force F_c : Force exerted on the control device.

Application force F_s : On friction brakes, the total force applied to a brake lining and which causes the braking force by the effect of friction.

Braking torque: Product of frictional forces resulting from the application force and the distance between the points of application of these forces and the axis of rotation of the wheels.

Total braking force F_f : Sum of the braking forces at the tire contact patches of all the wheels and the ground, produced by the effect of the brake system, and which oppose the movement or the tendency of the vehicle to move.

Braking-force distribution: Specification of braking force according to axle, given in % of the total braking force F_f . Example: front axle 60 %, rear axle 40 %.

Brake coefficient C^* : Defines the relationship between the total peripheral force of a given brake and the brake's application force.

$$C^* = \frac{F_u}{F_s}$$

F_u Total peripheral force,
 F_s Application force.

The mean is employed when there are variations in application forces at individual brake shoes (i number of brake shoes):

$$F_s = \sum \frac{F_{si}}{i}$$

Time periods

Reaction time (see Figure 1): The time that elapses between perception of the state or object which induces the response, and the point at which the control device is actuated (t_0).

Actuating time of the control device: Elapsed time between the moment when the component of the control device (t_0) on which the control force acts starts to move, and the moment when it reaches its final position corresponding to the applied control force (or its travel). (This is equally true for application and release of the brakes).

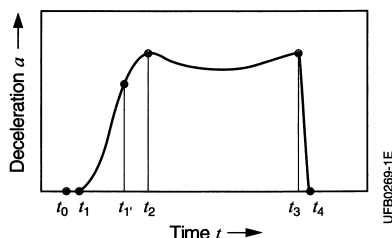
Initial response time $t_1 - t_0$: Elapsed time between the moment when the component of the control device on which the control force acts starts to move and the moment when the braking force takes effect.

Pressure build-up time $t_1' - t_1$: Period that elapses between the point at which the braking force starts to take effect and the point at which a certain level is reached (75 % of asymptotic pressure in the wheel-brake cylinder as per EU Directive 71/320/EEC [3], Annex III/2.4).

Initial response and pressure build-up time: The sum of the initial response and pressure build-up times is used to assess how the brake system behaves over time until the moment at which full braking effect is reached.

Active braking time $t_4 - t_1$: Elapsed time between the moment when the braking force starts to take effect and the moment when the braking force ceases. If the

Figure 1: Times and deceleration during braking to a stop



before t_0 : Reaction time,
 t_0 : Initial application of force on control device,
 t_1 : Start of deceleration,
 t_1' : End of pressure build-up time,
 t_2 : Fully developed deceleration,
 t_3 : End of maximum retardation,
 t_4 : End of braking operation (vehicle stationary),
 $t_1 - t_0$: Initial response time,
 $t_1' - t_1$: Pressure build-up time,
 $t_3 - t_2$: "Mean maximum retardation" range,
 $t_4 - t_1$: Active braking time,
 $t_4 - t_0$: Total braking time.

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vehicle stops before the braking force ceases, the time when motion ceases is the end of the active braking time.

Release time: Elapsed time between the moment when the control device starts to release and the moment when the braking force ceases.

Total braking time $t_4 - t_0$: Elapsed time between the moment when the control device on which the control force acts starts to move and the moment when the braking force ceases. If the vehicle stops before the braking force ceases, the time when motion ceases is the end of the active braking time.

Braking distance s

Distance traveled by the vehicle during the total braking time. If the time when motion ceases constitutes the end of the total braking time, this distance is called the "stopping distance".

Braking work W

Integral of the product of the instantaneous braking force F_t and the elementary movement ds over the braking distance s :

$$W = \int_0^s F_t ds.$$

Instantaneous braking power P

Product of the instantaneous total braking force F_t and the vehicle's road speed v :

$$P = F_t v.$$

Braking deceleration

Reduction of speed obtained by the brake system within the considered time t . A distinction is made between the following:

Instantaneous braking deceleration

$$a = \frac{dv}{dt}.$$

Mean braking deceleration over a period of time

The mean braking deceleration between two points in time t_B and t_E is

$$a_{mt} = \frac{1}{t_E - t_B} \int_{t_B}^{t_E} a(t) dt;$$

This means that:

$$a_{mt} = \frac{v_E - v_B}{t_E - t_B},$$

where v_B and v_E are the vehicle speeds at the times t_B and t_E .

Mean braking deceleration over a specific distance

The mean braking deceleration over the distance between two points s_B and s_E is:

$$a_{ms} = \frac{1}{s_E - s_B} \int_{s_B}^{s_E} a(s) ds;$$

This means that:

$$a_{ms} = \frac{v_E^2 - v_B^2}{2(s_E - s_B)}$$

where v_B and v_E are the vehicle speeds up to the points s_B and s_E .

Mean braking deceleration over the total braking distance

The mean braking deceleration is calculated according to the equation:

$$a_{ms0} = \frac{-v_0^2}{2s_0},$$

where v_0 relates to the time t_0 (special instance of a_{ms} where $s_E = s_0$).

Mean fully developed deceleration d_m

Mean fully developed deceleration over the distance determined by the conditions $v_B = 0.8 v_0$ and $v_E = 0.1 v_0$ thus:

$$d_m = \frac{v_B^2 - v_E^2}{2(s_E - s_B)}$$

The mean fully developed deceleration is used in ECE Regulation 13 [6] as a measure of the effectiveness of a brake system. Since positive values for d_m are used here, the mathematic sign has been reversed in this case. (In order to establish a relationship between braking distance and braking deceleration, braking deceleration must be expressed as a function of the distance traveled.)

Braking factor z

Ratio between the total braking force, F_t , and the permissible total static weight, G_s , exerted on the axle or axles of the vehicle:

$$z = \frac{F_t}{G_s}.$$

Legal regulations

General Certification for a vehicle with regard to its brake system may only be granted when the brake system complies with the following regulations:

- §41 StVZO [4] (German Road Vehicle Registration Regulation) in conjunction with §72 StVZO [5] and the associated directives.
- Council Directive of the European Community (RREG) 71/320/EEC [3], associated Amending Directives and Annexes.
- ECE Regulations R13 [6], R13H [7], R78 [8].

In §41 StVZO, the requirements placed on the brake system differ depending on the type, gross weight rating, application, date of registration, and type-determined top speed. In the EC Directives, the requirements are allocated to individual vehicle categories. The vehicle categories are as follows:

- M1, M2, M3: Passenger cars with at least four wheels,
- N1, N2, N3: Commercial vehicles with at least four wheels,
- O1, O2, O3, O4: Trailers and semitrailers,
- L1, L2, L3, L4: Motorcycles, threewheelers.

The values stipulated in §41 StVZO with regard to mean fully developed deceleration do not apply, for example, to the recurring inspections required in Germany for registered vehicles on the road (general inspection, safety inspection). In these inspections, the requirements of §29 StVZO [9], s. 1, Annex VIII in conjunction with Annex VIIIa, Guideline for performing the general inspection and Guideline for performing the safety inspection, apply.

The requirements in §41 StVZO and ECE-R13H with regard to brake equipment are essentially identical. However, ECE Regulations R13, R13H and R78 have been further updated and also contain, for example, regulations covering electronically controlled brake systems. The prescribed braking effects must be determined in accordance with Directive 71/320/EEC, s. 1.1.2, Annex II, amended

by Directive 98/12/EC [10], or in accordance with §41 StVZO, s. 12.

Requirements placed on brake systems

(as per §41 StVZO, EC Directive 71/320/EEC, ECE-R13, as at 2009)

Category M and N vehicles must comply with the provisions which pertain to the service, secondary and parking brake systems. The brake systems may have common components. Such vehicles must have at least two mutually independent control devices for the brake systems, one of which must be lockable. The control devices must be fitted with separate transmission devices, each must be able to continue operating when the other fails. The distribution of braking force between the individual axles is prescribed and must be sensible. If a malfunction occurs, it must be possible to achieve the prescribed secondary braking effect with the remaining operational part of the brake system or with the vehicle's other brake system without the vehicle departing from its lane.

Vehicles in categories M2 and N2 and above must be fitted with automatic anti-lock braking facilities. Regulation EC 661/2009 [11] stipulates that from 1 November 2011 all new vehicle models and from 2014 all new category M1 and N1 vehicles to be brought onto the road must be equipped with an electronic driving-dynamics control system (Electronic Stability Program). This regulation also applies, with the exception of off-road vehicles as per Directive 2007/46/EC [12], Annex II, Part A, to vehicles in the following categories:

- M2 and M3, except vehicles with more than three axles, articulated buses and buses of category 1 or A,
- N2 and N3, except vehicles with more than three axles, tractor units with a total weight of between 3.5 and 7.5 t and special-purpose vehicles as per Directive 2007/46/EC, Annex II, Part A,
- O3 and O4, with pneumatic suspension, except vehicles with more than three axles, trailers for heavy transports and trailers with areas for standing passengers.

The implementation of this regulation for the vehicle categories, except categories M1 and N1, is defined in Regulation EC 661/2009 [11], Annex V.

Continuous-operation brake systems

Continuous-operation brake systems are additionally used to relieve the strain on the service brake on long downhill gradients. Category M3 vehicles for local and long-distance duty (buses weighing more than 5.5 t (except city buses)) and other category N2 and N3 vehicles with a gross weight rating of more than 9 t (Directive 71/320/EEC, §41 StVZO, s. 15) must be equipped with a retarder. Exhaust brakes or similar facilities are classed as retarders. The retarder must be designed to hold the vehicle when fully laden when driving on a downhill gradient of 7% and a distance of 6 km at a speed of 30 km/h.

Category O trailers

Category O1 trailers are not required to have their own brake system; a securing connection to the tractor vehicle is sufficient. Category O2 trailers and above must be fitted with service and parking brake systems, which may have common components. The distribution of braking force between the individual axles is prescribed in Directive 71/320/EEC. It must be sensibly distributed between the axles.

Category O3 trailers and above (ECE), as well as trailers and semitrailers with a gross weight rating of more than 3.5 t and a type-determined top speed of more than 60 km/h (§41b StVZO, s. 2), must be equipped with an antilock braking facility. Semitrailers only need to be equipped with ABS if the gross weight rating reduced by the fifth-wheel load exceeds 3.5 t.

Category O3 trailers (existing types) which are registered for use on public roads after 11.07.2014 or 01.11.2014 must be equipped with an electronic driving-dynamics control system (Electronic Stability Program). For new types, this regulation will already apply from 01.11.2011 or 11.07.2012 (Regulation EC 661/2009 [11], Annex V).

Category O2 trailers and below must be equipped with inertia brake systems. The trailer must brake automatically if it be-

comes decoupled from the tractor vehicle while moving, or (for trailers weighing less than 1.5 t) it must be equipped with a securing connection to the tractor vehicle.

Category L vehicles

Motorized two- and three-wheeled vehicles must be equipped with 2 mutually independent brake systems. In the case of duty category L5 three-wheel vehicles, the two brake systems must both act on all the wheels. A parking brake system must be fitted.

Tractor vehicles and trailers with compressed-air brake systems

The compressed-air connections between the individual vehicles must be of the dual- or multi-line design. This ensures that the compressed-air brake system in the trailer can also be refilled during the braking operation. When the service brake system on the tractor unit is operated, the service brake system on the trailer must also be operated with a variable degree of force. If a fault occurs in the service brake system of the tractor unit, that part of the system not affected by the fault must be capable of braking (controlling) the trailer with a variable degree of force. If one of the connecting lines between tractor unit and trailer is interrupted or develops a leak, it must still be possible to brake the trailer, or it must brake automatically.

The energy accumulators of the service brake systems must be designed such that, after eight full operations of the service brake, at least the requested secondary braking effect is furnished on the ninth braking. The energy accumulators must not be replenished during this test. The braking effect of the individual vehicles is prescribed as a function of the pressure at the "Brake" coupling head in Directive 71/320/EEC.

Vehicles with antilock braking facilities

Antilock braking facilities must comply with Directives 71/320/EEC, Annex X and ECE-R13, Annex 13. An antilock braking facility is part of a service brake system which automatically controls the slip in the direction of wheel rotation at one or more

of the vehicle's wheels during the braking operation. The requirements placed on the antilock braking facility differ, depending on the category, for vehicles ABS categories 1, 2 and 3, for trailers ABS categories A and B.

Essential requirements placed on the antilock braking facility (Category 1) are:

- Locking of the directly controlled wheels under braking must be prevented at speeds of over 15 km/h on all road surfaces.
- Directional stability and maneuverability must be maintained. Under a μ -split condition (extremely different coefficients of friction between the left and right wheels), steering corrections of 120° during the first two seconds and 240° in total are permissible.
- There must be a special visual warning system (yellow warning signal) to indicate electrical faults.
- Motor vehicles (except Categories M1 and N1) with ABS which are equipped to tow a trailer with ABS must be fitted with a separate visual warning system (yellow warning signal) for the trailer. Transfer must be effected via pin 5 of the electrical plug-in connection as per ISO 7638 [13].
- The energy accumulators of the service brake system in vehicles with ABS must be designed such that the prescribed secondary braking effect is still achieved even after a controlled braking operation of longer duration ($t = v_{\max}/7$, at least 15 seconds) and then four uncontrolled full-braking operations without energy replenishment.

Requirements and test conditions

The required values and test conditions must be applied during the test in accordance with § 19 [14] and § 20 StVZO [15]. Departures from the test method described in Directive 71/320/EEC, Annex II, s. 1.1.2, last amended by Directive 98/12/EC and the test method specified in § 41 StVZO, s. 12, are permitted – especially during verification checks as per § 29 StVZO – if the condition and the effect can be ascertained by other means (§ 41 StVZO, s. 12). When vehicles to be newly registered are tested, a higher braking deceleration corresponding to the usual

easing of the braking effect must be achieved; furthermore, sufficient state-of-the-art continuous brake duty must be guaranteed for longer stretches of downhill driving.

Minimum retardation and max. permissible control forces during the general inspection as per § 29 StVZO

The required retardation values are determined on test benches and only in exceptional cases in road tests. The required values are maximum values, because the time response which is needed to determine the mean fully developed deceleration is not measured in these recurring inspections.

Structure and organization of brake systems

Essential requirements

The brake systems in motor vehicles must comply with the requirements of Directive 71/320/EEC, ECE-R13 Part 1, ECE-R13 Part 2 and ECE-R13 H, and other country-specific regulations. Motor vehicles must be equipped with two separate brake systems, one of which must be lockable. The brake systems must have separate control devices. In the event of a fault in the service brake system, it must still be possible for at least two wheels (not on the same side) to be braked.

Types of brake system

The brake systems comprise the service brake and parking brake systems and (in commercial vehicles and motor buses) the continuous-operation (retarder) brake system. The requested secondary brake system normally comes into play when a fault occurs in the service brake system. Special-purpose vehicles with special requirements may also have special braking functions such as a hill-climbing brake or an anti-jackknifing brake.

Type of force generation

When it comes to how the force is generated, there are three different types of system: muscular-energy, energy-assisted, and non-muscular-energy brake systems. In muscular-energy systems, the muscular force of the driver alone is effective, in energy-assisted systems this is boosted by booster systems (brake booster), and in non-muscular-energy systems, the driver's control force acts only as a control variable. The maximum required control forces are prescribed for each type of vehicle.

Transmission device

Force is transmitted from the control device to the wheel brakes by mechanical, hydraulic, pneumatic or electric means. Mechanical force transmission is only customary and prescribed for parking brake systems (§ 41 StVZO, s. 5).

Force transmission for the service brake system is performed via two separate brake circuits by hydraulic or pneumatic

means so that at least one brake circuit remains in operation in the event of a fault.

Electrical brake operation has up to now only been used in electrically acting parking brake systems (see Electromechanical parking brake systems).

Brake-circuit configuration

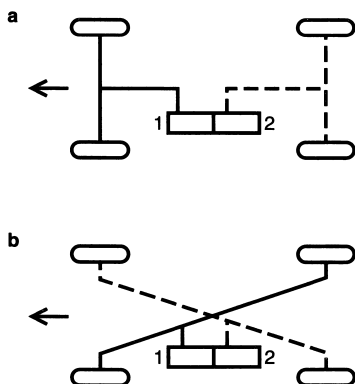
Configuration of the brake circuits is governed by DIN 74000 [16]. In category M1 vehicles (passenger cars), the brake-circuit configuration is often diagonal (Figure 2b). But this is only possible in conjunction with suitable front-axle geometry (steering offset negative or neutral). In all other vehicle categories, the II configuration is used (Figure 2a). Here the front axle forms one of the brake circuits and the rear axle forms the other. All other brake-circuit configurations as per DIN 74000 are rarely used today and are therefore no longer described in these pages. The direct demand for a dual-circuit design of the transmission device is prescribed in § 41 StVZO, s. 16 only for motor buses.

Figure 2: Variants of brake-circuit configuration

- a) II configuration,
b) X configuration.

- 1 Brake circuit 1,
2 Brake circuit 2.

← Direction of travel.



Braking-force distribution

Directives 71/320/EEC, ECE-R13 and ECE-R13H also place requirements on braking-force distribution between the individual axles. This must be sensibly distributed in all load states between the axles. Braking-force distribution can be effected on the one hand through an assembly-related configuration of the wheel brakes and on the other hand through a vehicle-related configuration. Among other things, the center-of-gravity height, the wheel-base and the empty-empty ratio of the vehicle are taken into consideration.

In commercial vehicles, according to the diagrams in Directive 71/320/EEC, braking-force distribution is also dependent on the pressure at the "Brake" coupling head. Vehicle-related configuration of braking-force distribution is effected by the integration of a braking-force limiter or an automatically acting braking-force metering device (automatic load-sensitive braking-force metering).

In modern vehicles braking-force distribution is integrated as an additional function into the electronic wheel-slip control system (antilock braking facility, driving-dynamics control).

Assemblies

Brake systems in motor vehicles consist of the following assemblies, which differ in design depending on whether the system is hydraulic or pneumatic: energy supply, control devices, transmission devices, control facilities, wheel brakes, and auxiliary devices.

References

- [1] ISO 611: Road vehicles – Braking of automotive vehicles and their trailers – Vocabulary.
- [2] DIN 70024: Vocabulary for components of motor vehicles and their trailers.
- [3] EC Directive 71/320/EEC: Council Directive of 26 July 1971 on the approximation of the laws of the Member States relating to the braking devices of certain categories of motor vehicles and their trailers.
- [4] §41 StVZO: Brakes and wheel chocks.
- [5] §72 StVZO: Entry into force and transitional provisions.

[6] ECE-R13: Standard conditions for approval of category M, N and O vehicles with regard to brakes.

[7] ECE-R13H: Standard conditions for approval of passenger cars with regard to brakes. Day of entry into force: 11 May, 1998.

[8] ECE-R78: Standard conditions for approval of category L1, L2, L3, L4 and L5 vehicles with regard to brakes.

[9] §29 StVZO: Inspection of motor vehicles and trailers.

[10] EC Directive 98/12/EC of the Commission of 27 January 1998 adapting to technical progress Council Directive 71/320/EEC on the approximation of the laws of the Member States relating to the braking devices of certain categories of motor vehicles and their trailers.

[11] Regulation (EC) No. 661/2009 of the European Parliament and of the Council of 13 July 2009 concerning type-approval requirements for the general safety of motor vehicles, their trailers and systems, components and separate technical units intended therefore.

Published in the Official Journal of the European Union L200 of 31. July 2009.

[12] Directive 2007/46/EC of the European Parliament and of the Council of 5 September 2007 establishing a framework for the approval of motor vehicles and their trailers, and of systems, components and separate technical units for such vehicles (Framework Directive).

[13] ISO 7638: Road vehicles – Connectors for the electrical connection of towing and towed vehicles.

[14] §19 StVZO: Granting and effectiveness of design certification.

[15] §20 StVZO: General Certification for types.

[16] DIN 74000: Hydraulic braking systems; dual circuit brake systems; symbols for brake circuits diagrams.

Brake systems for passenger cars and light utility vehicles

Subdivision of passenger-car brake systems

Brake systems for passenger cars and light utility vehicles must comply with the requirements of various directives and statutory provisions, e.g., 71/320/EEC [1], ECE R13 [2], ECE R13-H [3] and in Germany §41 StVZO [4]. The requirements with regard to functioning, effect and test methods are set out in these regulations.

The entire system is subdivided into the service-brake system, the parking-brake system and the secondary-brake system.

Service-brake system

The service-brake system allows the driver to reduce with graduable effect the speed of a vehicle during normal driving or to bring the vehicle to a halt. In passenger cars and light utility vehicles it is normally designed as an energy-assisted brake system.

The driver meters the braking effect steplessly by pressing on the brake pedal.

Force is transmitted to the wheel brakes via the tandem brake master cylinder to two mutually independent hydraulic transmission devices (Figure 1). The service-brake system acts on all four wheels.

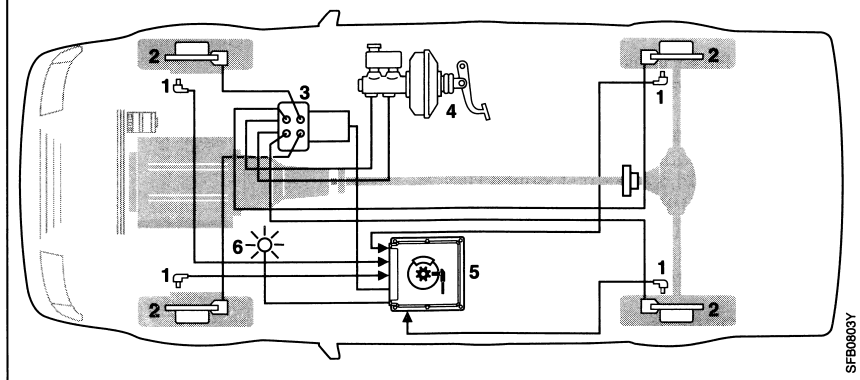
Parking-brake system

The parking-brake system ("handbrake") is an independent brake system which holds the vehicle stationary, even on a gradient and especially when the driver is not in the vehicle. The holding-stationary mechanism is integrated in the wheel brake. Legal requirements stipulate that the parking brake must have a consistently mechanical connection between the control device and the wheel brake, e.g., by means of a linkage or a control cable.

The parking brake is usually actuated by a handbrake lever next to the driver's seat, and in some cases also by a pedal. In the case of electrically actuated parking-brake systems, the parking brake is locked or released by means of an electrical operating device (switch). The

Figure 1: Hydraulic dual-circuit brake system

- 1 Wheel-speed sensors,
- 2 Wheel brakes (disk brakes, drum brakes also possible on the rear axle),
- 3 Hydraulic modulator (for antilock braking system and driving-dynamics control system),
- 4 Control device with brake booster, tandem brake master cylinder and expansion reservoir,
- 5 ECU (can be directly mounted on the hydraulic modulator),
- 6 Warning lamp for antilock braking system and driving-dynamics control.



service- and parking-brake systems are thus equipped with separate, individual control and transmission devices. The parking-brake system may be graduable in design, acting on the wheels of only one axle.

The holding-stationary effect is calculated according to ECE R13-H on a downhill gradient with fully laden vehicles. The downhill gradient for single vehicles is 20 %. If the vehicle is equipped to tow a trailer, the holding-stationary effect must also be achieved with an unbraked trailer on a downhill gradient of 12 %.

Secondary-brake system

In the event of a fault, e.g., leak or fractured pipe, it must still be possible with the operational part of the brake system to achieve at least the secondary braking effect – with the identical control force at the control device. It must be possible to meter the secondary braking effect, which in turn must be at least 50 % (ECE R13-H) or 44 % (§41 s. 4a). The vehicle must not leave its lane when the secondary brake is applied.

The secondary-brake system does not need to be an independent third brake system (in addition to the service- and parking-brake systems) with a special control device. Either the intact brake circuit of a dual-circuit service-brake system or a graduable parking-brake system can be used as the secondary-brake system.

Components of the passenger-car brake system

Control device

The control device comprises those parts of the brake system which initiate the effect of this brake system. When the service brake is applied, the force of the driver's foot acts on the brake pedal. The lever-transmitted pedal force is boosted in the brake booster, depending on its design, by a factor of 4 to 10 and acts on the piston in the brake master cylinder (Figure 1). The control force is converted into hydraulic pressure. Under full braking, this pressure ranges between 100 and 160 bar, depending on the system configuration.

Vacuum brake booster

Function

The brake booster reduces the applied control force required for the braking operation, but must not impair the sensitive graduation of the braking force and the feeling of the measure of braking.

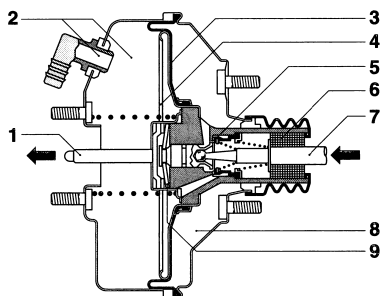
Design

Brake boosters function as vacuum-operated boosters or also hydraulically. Hydraulic brake boosters are supplied by the power steering or by a separate hydraulic pump and pressure-accumulator devices.

Passenger-car brake systems are usually equipped with vacuum brake boosters. These vacuum boosters utilize the negative pressure generated in the intake manifold during the induction stroke on gasoline engines or the vacuum (0.5 – 0.9 bar) produced by a vacuum pump on

Figure 2: Vacuum brake booster

- 1 Push rod,
- 2 Vacuum chamber with vacuum connection,
- 3 Diaphragm,
- 4 Power piston,
- 5 Valve unit,
- 6 Air filter,
- 7 Piston rod,
- 8 Working chamber,
- 9 Reaction element.



diesel engines and on electric vehicles or hybrid vehicles to amplify the force applied by the driver's foot. A diaphragm separates the vacuum chamber with vacuum connection from the working chamber (Figure 2). The piston rod transmits the applied foot pressure to the working piston and the amplified force is passed to the brake master cylinder via the push rod.

Operating principle

If the brake is not operated, the vacuum chamber and working chamber are connected via the valve unit. Given that the vacuum connection is connected to a vacuum source, this means that there is a vacuum in both chambers.

As soon as a braking operation is initiated, the piston rod moves forwards in the direction of the arrow. After a short stroke the connection between the working chamber and the vacuum chamber is blocked. As the piston rod continues to move, the inlet valve in the valve unit is opened and atmospheric air flows into the working chamber. The pressure in the working chamber is then greater than in the vacuum chamber. The atmospheric pressure acts via the diaphragm on the diaphragm disk with which it is in contact. Because the diaphragm disk is attached to the valve unit, the latter moves when the disk moves, thereby assisting the foot pressure transmitted by the connecting rod. Maximum boost is dependent on the effective diaphragm or piston area, on the atmospheric pressure and on the effective vacuum.

When the braking operation has ended, the inlet valve is closed and the vacuum and working chambers are connected via the valve unit. In this way, the pressure (vacuum) is identical in both chambers.

Vacuum non-return valve

In all brake systems which have a vacuum brake booster a non-return valve is incorporated in the vacuum line between vacuum source and brake booster. While there is a vacuum present, the non-return valve remains open. It closes when the vacuum source ceases to produce a vacuum (engine is switched off) so that the vacuum inside the brake booster is

retained. Thus, brake boost is effective even when the engine is switched off for several brake actuations.

Electromechanical brake booster

The electromechanical brake booster referred to as iBooster by Bosch satisfies, thanks to its electronic control, new demands imposed on brake systems. These demands include for example lower or no availability of vacuum in the vehicle, reduced CO₂ emissions, and redundancy for highly automated driving. The iBooster can be used with all drive concepts, including hybrid and electric vehicles. Like the vacuum brake booster, the iBooster supports the driver with an assisting force (electric motor via transmission).

Operating principle

The iBooster (Figure 3) detects the driver's brake request via an integrated differential-travel sensor and sends this information to the ECU. The ECU calculates the activation of the electric motor, which converts its torque via a gear unit into the requested assisting force. Here the electric motor is activated in such a way that the differential travel between the input rod connected to the brake pedal and the transmission element connected to the electric motor is compensated to zero. The sum total of the force supplied by the booster and the driver is converted into hydraulic pressure in a standard brake master cylinder. The iBooster's resulting pedal characteristic is dependent on the design of the components (e.g., on the maximum motor force). Certain parameters can, in contrast to the vacuum brake booster, be additionally influenced by the software logic.

Special features

The iBooster enables the pedal characteristic to be adapted whereby the assisting force is adapted by means of a change to the target-value calculation of the control facility. It is thus possible to adapt the so-called jump-in and the boost factor (Figure 4) within certain limits to the requirements specified by the vehicle manufacturer. Jump-in is the point at which the driver force proportionally influences the brake-booster braking force. Below the

jump-in the braking force comes solely from the booster itself. The driver must first overcome spring forces before his/her initiated force influences the braking force.

In vehicles with an electric or hybrid drive the iBooster, in combination with a special type of driving-dynamics control (Electronic Stability Control ESC), enables brake energy to be recuperated to a vehicle deceleration of 0.3 g without affecting the braking feel. Here during braking the decelerations caused by wheel brake and electric machine are variably matched to each other without additional components. In hybrid vehicles this regenerative braking reduces fuel consumption and CO₂ emissions – especially in the event of frequent braking and accelerating in urban traffic.

The iBooster can build up brake pressure automatically (without actuating the brake pedal) with the aid of the engine/transmission unit. Compared with typical driving-dynamics control systems, the required brake pressure is built up faster and adjusted with greater accuracy. This is beneficial for example for automatic emergency-braking systems and ACC functions.

In combination with driving-dynamics control the iBooster offers the brake-system redundancy required for automated driving. Both systems are, independently of one another, able to generate brake pressure and decelerate the vehicle.

Brake master cylinder

The brake master cylinder converts the foot force applied by the driver and boosted by the brake booster into hydraulic brake pressure.

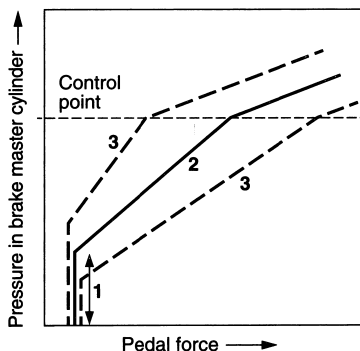
Brake master cylinder with central valve

Design

In order to comply with statutory safety requirements, service-brake systems are

Figure 4: Adaptation of pedal characteristic

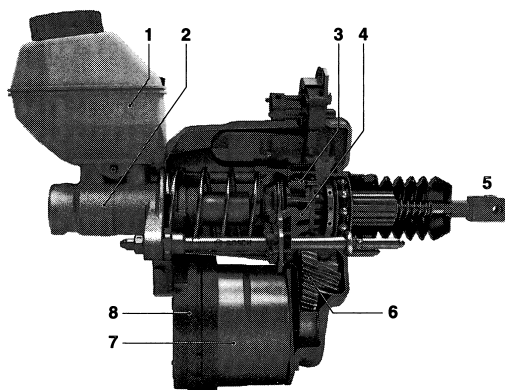
- 1 Jump-in,
- 2 Pedal characteristic resulting from design,
- 3 Adaptation of pedal characteristic by software (adaptation of jump-in and adaptation of iBooster assisting force).



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Figure 3: Electromechanical brake booster (iBooster)

- 1 Brake-fluid reservoir,
- 2 Brake master cylinder,
- 3 Differential-travel sensor,
- 4 Transmission element,
- 5 Input rod,
- 6 Transmission,
- 7 Electric motor,
- 8 Control unit.



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equipped with two separate service-brake circuits. If a leak occurs (circuit failure), the other circuit remains intact (secondary braking effect). This can be achieved by a tandem brake master cylinder (Figure 5). The compression spring of the intermediate-piston circuit in the rest state holds the intermediate piston and the push-rod piston against the rear stop. The compensating port and the central valve are opened. Both hydraulic service-brake circuits are depressurized (drive position).

Operating principle

The force applied at the brake pedal and boosted by the brake booster acts directly

on the push-rod piston and pushes it to the left. After short piston travel the compensating port is sealed and a pressure build-up can take place in the push-rod circuit. This also causes the intermediate piston to be pushed to the left.

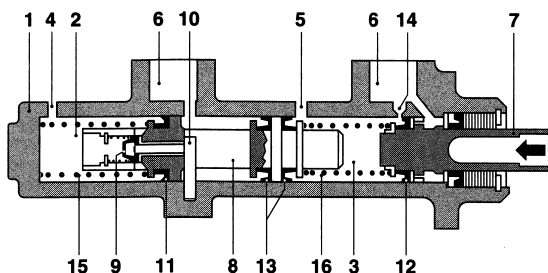
Brake master cylinder with captive piston spring

Design

The "captive" piston spring – a compression spring – in the rest state holds the push-rod piston and the intermediate piston always at the same distance (Figure 6). This prevents the piston spring in the rest state from pushing the inter-

Figure 5: Tandem brake master cylinder with central valve in the intermediate-piston circuit

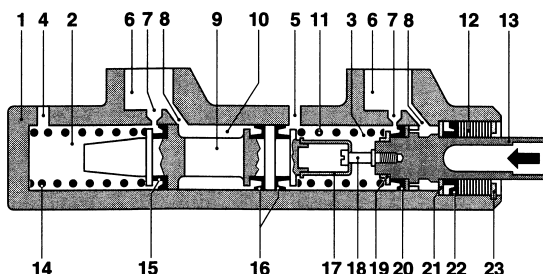
- 1 Cylinder housing, 2 Pressure chamber, intermediate-piston circuit,
- 3 Pressure chamber, push-rod circuit, 4 Pressure port, intermediate-piston circuit,
- 5 Pressure port, push-rod circuit, 6 Connection for brake-fluid reservoir, 7 Push rod,
- 8 Intermediate piston,
- 9 Central valve,
- 10 Stop for central valve,
- 11 Primary cup seal, intermediate piston,
- 12 Primary cup seal, push-rod piston,
- 13 Separating cup seal,
- 14 Compensating port,
- 15 Compression spring, intermediate-piston circuit,
- 16 Compression spring, push-rod circuit.



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Figure 6: Tandem brake master cylinder with captive piston spring

- 1 Cylinder housing, 2 Pressure chamber, intermediate-piston circuit,
- 3 Pressure chamber, push-rod circuit, 4 Pressure port, intermediate-piston circuit,
- 5 Pressure port, push-rod circuit, 6 Connection for brake-fluid reservoir, 7 Compensating port,
- 8 Replenishing port, 9 Intermediate piston, 10 Space, 11 Captive piston spring,
- 12 Plastic bush, 13 Push-rod piston,
- 14 Compression spring, intermediate-piston circuit,
- 15 Primary cup seal, intermediate piston,
- 16 Separating cup seal,
- 17 Stop sleeve,
- 18 Stop screw,
- 19 Support ring,
- 20 Primary cup seal, push-rod piston,
- 21 Stop disk,
- 22 Secondary cup seal,
- 23 Snap ring.



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mediate piston and the latter from over-running the compensating port with the primary cup seal. In this situation pressure compensation via the compensating port would no longer be possible in the secondary circuit, and in the event of a residual pressure the brake shoes would not lift off the brake drums when the brake releases.

Operating principle

When the brake is actuated, the push-rod piston and the intermediate piston move in the direction of the arrow to the left, overrun the compensating ports and force brake fluid through the pressure ports into the brake circuits. As pressure increases the intermediate piston is no longer moved by the captive piston spring, but instead by the pressure of the brake fluid.

Expansion reservoir

The expansion reservoir, also called the brake-fluid reservoir, is mounted directly on the brake master cylinder and connected to it via two ports. It is both the reservoir for the brake fluid and the expansion reservoir. It compensates volume fluctuations in the brake circuits which occur after the brake is released, in response to wear of the brake linings and to temperature differences in the brake system, and during intervention by the antilock braking system (ABS) or the driving-dynamics control (Electronic Stability Control).

Transmission device

The hydraulic pressure is transmitted by the brake fluid via brake pipes as per DIN 74234 [5] and brake hoses as per SAE J 1401 [6] to the wheel-brake cylinders. Brake fluids must comply with the requirements laid down in SAE J 1703 [7] or FMVSS 116 [8] (see Brake fluids).

Wheel brakes

Floating-caliper disk brakes are usually used on the front wheels, but fixed-caliper disk brakes may also be used. Both floating-caliper disk brakes with integrated locking mechanism and Simplex drum brakes are used on the rear wheels (see Wheel brakes). Combinations of disk brakes and drum brakes (drum-in-head

systems) can also be used on the rear wheels. In this case, the drum brake accommodated in the brake-disk chamber is used exclusively for the parking-brake system.

The parking-brake control device can be mechanically designed as a hand-brake lever or a footbrake pedal with locking mechanism. Force is generally transmitted via cables or linkage to the wheel brakes on the rear axle. In the case of electromechanical parking brakes, the brake is actuated by means of electric motors and gearings (see Electromechanical parking-brake system).

Hydraulic modulator

Arranged between the brake master cylinder and the wheel brakes is the hydraulic modulator of the antilock braking system or the driving-dynamics control system and, depending on the scope of functions, a braking-force regulator or a braking-force limiter. These components, by limiting and adapting the brake pressure mostly on the rear axle, ensure a sensible distribution of braking force between the front and rear axles. This function can, especially in vehicles with markedly different load states, be executed on a load-sensitive basis (automatic load-sensitive braking-force metering).

The hydraulic modulator modifies the brake pressure during the braking process in such a way that the wheels are prevented from locking. Depending on the control variation, this job is performed by several solenoid valves and an electrically driven supply pump. In passenger-car brake systems, the front axle is individually controlled, i.e., each wheel is braked according to the respective grip. The rear wheels are controlled according to the select-low principle so that both rear wheels are braked together according to the wheel which has the lower grip (see also Antilock braking system and Driving-dynamics control).

Electromechanical parking brake

System overview

Conventional parking-brake systems are muscular-energy brake systems and are operated by purely mechanical means via lockable hand levers or foot pedals or via

a crankgear. In electromechanical parking-brake systems, also referred to simply as electromechanical parking brakes or automatic parking brakes, the control (operating) force is generated by an electric drive.

Operation and control are effected electrically via a switch or via logic control commands by other ECUs, which enable automatic closing or opening of the parking brake. The electromechanical parking brake can only be operated when the vehicle is stationary or at low speeds (usually 3 to 15 km/h). This must also be possible when the ignition and starting switch is turned off. If electric parking-brake systems are operated at higher speeds, an emergency-braking operation is first executed by the driving-dynamics control system. The parking brake is closed when the stationary vehicle status is attained within this braking operation.

The application force in the parking mechanism (see Parking brake) depends on the slope of the gradient on which the vehicle was parked. For this purpose, depending on the system, a tilt sensor is installed in the ECU of the electromechanical parking brake or corresponding sensor signals from other ECUs are used (e.g., airbag or chassis control). The retensioning of the brake necessitated by the cooling of the mechanical brake components is performed preventatively or according to a calculated temperature model and after vehicle movement has been detected.

A safety concept must ensure that unintentional activation in both the release and the closing directions due to electrical faults is ruled out. Furthermore, intentional activation of the electromechanical parking brake (emergency braking, only necessary if the control device of the service-brake system is broken) must not give rise to critical driving situations. If the operating unit of the electromechanical parking brake is deliberately actuated on a permanent basis, the driving-dynamics control system takes over the task of braking the vehicle at a speed in excess of 10 km/h. This ensures an optimally safe braking operation even in critical road situations. The electromechanical parking brake is locked only after the ve-

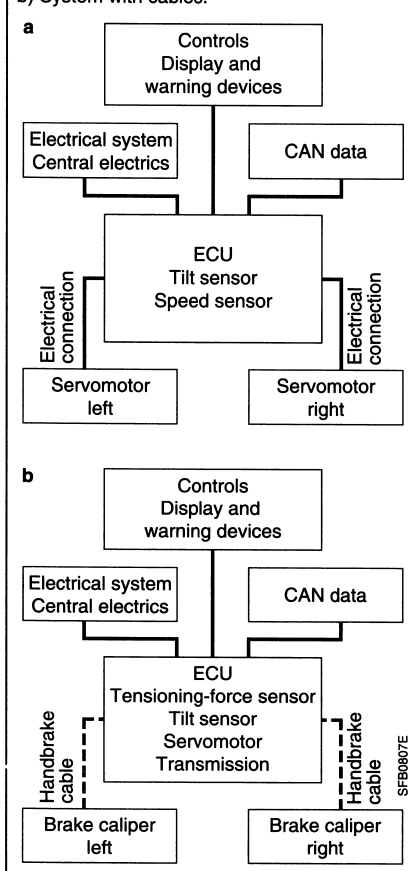
hicle speed has dropped below a specific threshold. The systems communicate with each other through an adequate data link (usually CAN or FlexRay).

Electric parking-brake systems may feature additional functions such as automatic release when starting.

Electric parking-brake systems are energy-assisted systems and are equipped with an emergency release device. It must also be possible for the system to be activated even when the ignition and starting switch is turned off, and the system may only be released when the ignition and starting switch is turned on and the brake

Figure 7: Electric parking-brake system

- a) System with servomotor on brake caliper,
b) System with cables.



pedal is simultaneously being pressed (or in the event of automatic release when the brake pedal is pressed).

The driver is always alerted to a locked parking brake system by a red warning lamp. Self-diagnosis detects malfunctions and faults, and indicates them via a warning lamp. A text message can also appear in a driver information display. The fault memory can be read out with a diagnostic tester and cleared after the fault has been corrected.

Diagnosis testers and relevant software may be required for servicing work, e.g., when replacing the brake pads.

Electromechanical parking brake with servomotor on the brake caliper

The electromechanical parking brake with servomotor comprises the following components (Figure 7a): operating unit, ECU, display and warning devices, tilt sensor (can be installed in the driving-dynamics control system), floating caliper with electric motor and multiple-stage gearing. A system distribution as described in VDA Recommendation 305-100 [9] is becoming increasingly common. This entails integrating the functionality of the parking brake in the driving-dynamics control system, thereby affording freedom in the choice of different manufacturers of these systems and the parking brake.

In the case of a brake caliper with an electric servomotor, force is transmitted for the parking-brake effect via a multiple-stage gearing and a threaded spindle. It is activated by way of an electrical switch (operating unit), which forwards the control commands to the ECU redundantly and in accordance with the safety concept. The ECU, taking into account further boundary conditions (e.g., road gradient), activates the electric servomotors via separate driver stages and electrical connecting leads.

A very high gear ratio means that very high application forces can be generated. These are in the region of 15 to 25 kN. As dictated by the concept, the electromechanical and hydraulic forces can be superimposed (superposition at the brake piston).

Electromechanical parking brake with cables

In the case of an electromechanical parking brake with cables, the following components are combined in a centrally arranged assembly – above the rear axle, in the passenger compartment or in the fender (Figure 7b): electric drive motor with gearing, required sensors (depending on the scope of functions, e.g., force, tilt, temperature and position sensors), ECU, and cable mechanism (if necessary with emergency release device).

This system too is activated by way of an electrical switch, which forwards the control commands to an ECU. The ECU activates the electric servomotor(s) via a driver stage. The application force can vary, depending on the road gradient. The system is automatically retensioned when the vehicle is stopped either after a cooling phase corresponding to a temperature model or after vehicle movement has been detected.

Electrohydraulic brake

Function

The electrohydraulic brake (EHB, “Sensotronic Brake Control”, SBC) is an electronic brake-control system with hydraulic actuator engineering. Like a conventional hydraulic brake, its function is to reduce vehicle speed, bring the vehicle to a halt, or keep the vehicle stationary. As an active braking system, it takes control of brake operation, braking-force boosting and braking-force control. Hydraulic standard wheel brakes are used as brakes.

Actuation unit

Mechanical operation of the brake pedal is detected by the actuation unit by means of electronic sensors with redundant backup (Figure 8). The pedal-travel sensor consists of two separate angle-position sensors. Together with the brake-pressure sensor for the pressure applied by the driver, this produces a threefold system for detecting driver input. The system can continue to function normally, even if one of the sensors fails.

The pedal-travel simulator produces an appropriate force/travel curve and calculates the amount of brake-pedal damping. The driver experiences the same “brake feel” with electrohydraulic brakes as with

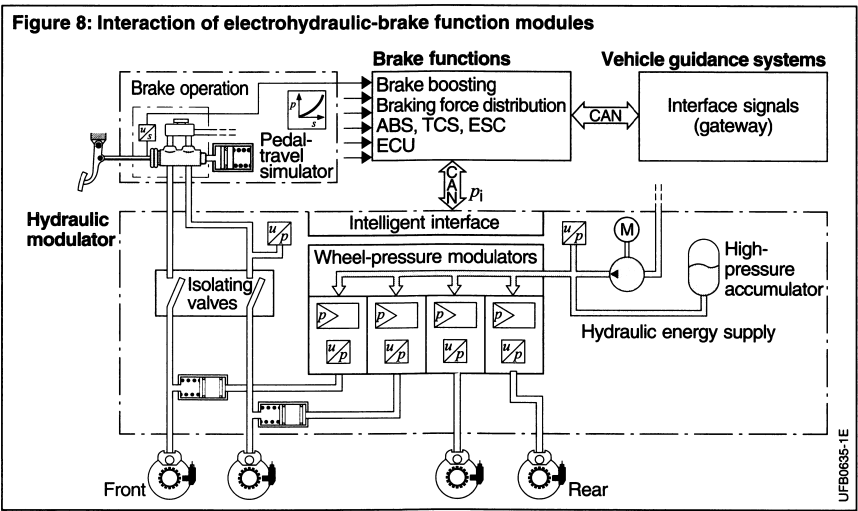
a very well designed conventional braking system.

A conventional brake booster is not required here. Only the driver’s brake request is determined in the actuation unit during normal operation; the brake pressure is generated in the hydraulic modulator. The brake master cylinder performs its function in the event of a system failure. The expansion reservoir supplies the hydraulic modulator with brake fluid.

Electronic control

The brake request is determined in the remote-mounted ECU from the sensor signals of the actuation unit. Braking characteristics can be adapted to the driving conditions (e.g., sharper response with more dynamic driving styles). A “duller” pedal characteristic can be used to alert the driver to a reduction in braking effect, when the brakes reach the limits of their effectiveness before overheating induces brake fading.

Further, the functions for the antilock braking system, the traction control system, and the driving-dynamics control are integrated in the ECU. In addition, comfort and convenience functions such as for example hill hold control, automatic brake-system prefilling when the foot is quickly removed from the accelerator pedal, the chauff-



feur brake (soft stop, jolt-free stopping by means of automated brake-pressure reduction shortly before standstill), and the brake-disk wiper are provided.

Thanks to complete electronic pressure control, the electrohydraulic brake can be easily networked with vehicle guidance systems (e.g., Adaptive Cruise Control, ACC).

Hydraulic modulator

Operating principle in normal operation

Figure 8 shows the electrohydraulic-brake components as a block diagram. An electric motor drives a hydraulic pump. This charges a high-pressure accumulator to a pressure of between approximately 90 and 130 bar, monitored by an accumulator pressure sensor. The four separate wheel-pressure modulators are supplied by the accumulator and set the required pressure at the wheel-brake cylinders separately for each wheel. The pressure modulators themselves each consist of two valves with proportional-control characteristics and a pressure sensor. Brake-pressure modulation and active braking are silent and generate no brake-pedal feedback.

In normal mode, the isolating valves isolate the brakes from the actuation unit. The system is in "brake-by-wire" mode. It electronically detects the driver's braking request and transmits it "by wire" to the wheel pressure modulators. The interaction of electric motor, valves, and pressure sensors is regulated by the ECU. This has two microcontrollers which monitor each another. The essential feature of these electronics is their extensive self-diagnosis, which monitors the plausibility of every system state at all times. It means that any faults can be displayed to the driver before a critical condition arises. If components fail, the system automatically provides the optimal remaining partial function to the driver.

An intelligent interface with the CAN bus provides the link between the remote-mounted ECU and the add-on of the hydraulic modulator.

Braking in the event of system failure

The electrohydraulic brake is designed so that in the event of serious faults (e.g.,

power-supply failure), it switches to a state in which the driver can brake the vehicle without using the active brake-booster function. When de-energized, the isolating valves establish a direct connection to the actuation unit and allow a direct hydraulic connection from the actuation unit to the wheel-brake cylinders (hydraulic fallback level).

References

- [1] 71/320/EEC: Council Directive of 26 July 1971 on the approximation of the laws of the Member States relating to the braking devices of certain categories of motor vehicles and their trailers.
- [2] ECE R13: Regulation No. 13 of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of vehicles of categories M, N and O with regard to braking.
- [3] ECE R13-H: Regulation No. 13-H of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of passenger cars with regard to braking.
- [4] §41 StVZO (road traffic licensing regulations, Germany) – Brakes and wheel chocks.
- [5] DIN 74234: Hydraulic braking systems; brake pipes, flares.
- [6] SAE J 1401: Road Vehicle Hydraulic Brake Hose Assemblies for Use with Non-petroleum-Base Hydraulic Fluids.
- [7] SAE J 1703: Motor Vehicle Brake Fluid.
- [8] FMVSS 116: Federal Motor Vehicle Standard No. 116: Motor Vehicle Brake Fluids.
- [9] VDA Recommendation 305-100: Recommendation for integration of electric parking brakes control into the ESC system (Electronic Stability Control) with regard to the ESC (ESC assembly) and the brake caliper (brake assembly).
- [10] B. Breuer, K.H. Bill (Editors): *Bremsenhandbuch*. 5th Edition, Verlag Springer Vieweg, 2017.

Brake systems for commercial vehicles

System overview

Brake systems for commercial vehicles and trailers must satisfy the requirements of various regulations such as, for example, RREG 71/320 EEC and ECE R13 [2]. Essential functions, effects and test methods are set out in these regulations. The entire system is subdivided into the service-brake, parking-brake, secondary-brake systems, and continuous-action brake systems.

Service-brake system

Service-brake system, tractor vehicle

The service-brake system, designed as an energy-assisted brake system in commercial vehicles (Figures 1 and 2), can operate with compressed air or also with a combination of compressed air and hydraulics.

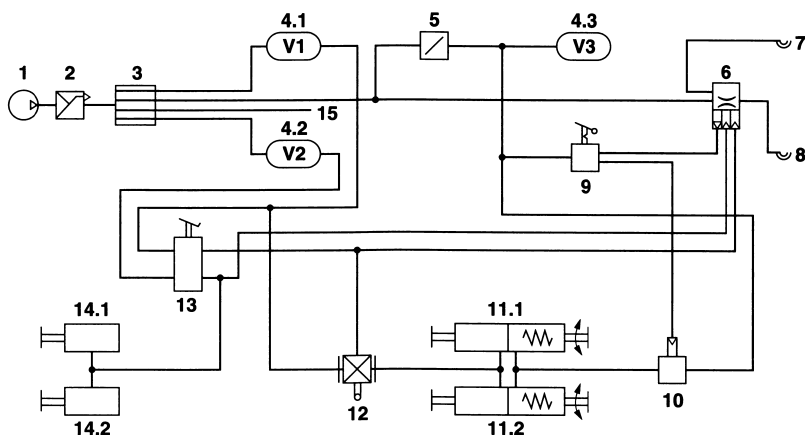
In the event of a fault, e.g. brake-circuit failure, it must still be possible with the op-

erational part of the system to achieve at least the secondary braking effect – with the identical control force at the normal control device. It must be possible to meter the effect, and the trailer must not be affected by this malfunction, i.e. the trailer control (trailer control valve) must have a dual-circuit design. The secondary braking effect must achieve at least 50 % of the braking effect of the service-brake system. It is therefore customary to split the system into two brake circuits already separate on the supply side, even though this configuration is only legally required in motor buses.

The energy supply to the trailer must be guaranteed even during the braking operation. The dual-line system became mandatory when RREG 71/320 came into force, but had already been available prior to this and was known by the name "Nato" brake.

Figure 1: Structure of a compressed-air brake system with trailer control

- 1 Air compressor, driven by engine, 2 Pressure regulator, 3 Four-circuit protection valve,
- 4.1 Air reservoir V1 for circuit 1, 4.2 Air reservoir V2 for circuit 2,
- 4.3 Air reservoir V3 for circuit 3 (trailer, pneumatic suspension), 5 Overflow valve with limited return flow, 6 Trailer control valve with throttle valve,
- 7 "Supply" coupling head (red), 8 "Brake" coupling head (yellow), 9 Parking-brake valve with test position, 10 Relay valve,
- 11.1 Combination brake cylinder, rear right, 11.2 Combination brake cylinder, rear left,
- 12 Load-dependent braking-force regulator (ALB), 13 Service-brake valve,
- 14.1 Brake cylinder, front right, 14.2 Brake cylinder, front left,
- 15 Secondary loads/ancillaries (e.g. pneumatic suspension, door-closing system).



The trailer is continually supplied with a defined pressure via the supply line. This pressure must be between 6.5 and 8.0 bar on an intact tractor vehicle, irrespective of the tractor vehicle's operating pressure established by the manufacturer. The trailer must be exchangeable. The trailer's service-brake system is controlled by a second line, the brake line. This line is also governed by the regulations pertaining to trailer exchangeability. Thus the pressure in the brake line must be 0 bar in driving mode, and 6.0 to 7.5 bar in fully-braked mode.

Service-brake system, trailer

The trailer has an independent service-brake system, which is only partly subject to the demand for a secondary braking effect. According to the requirements in RREG 71/320, the braking effects of the service-brake systems in the tractor vehicle and in the trailer must be within closely set tolerances as a function of the control pressure in the brake line to the trailer, i.e. they must be approximately the same (design tolerance band RREG 71/320 and ECE R13).

If the supply line or the brake line is fractured, the trailer must be able to be fully or partly braked or it must initiate an automatic braking. Commercial vehicles with electronically controlled brake systems have – in addition to the pneumatic

brake line – an electrical signal-transmission feature for electrically controlling the service-brake system in the trailer. This is

Figure 3: Compatibility diagram

Tractor vehicle and trailer RREG 71/320, ECE R13.

- T_R Sum total of braking forces at circumference of all trailer wheels,
- P_R Total static normal force of trailer,
- T_M Sum total of braking forces at circumference of all tractor-vehicle wheels,
- P_M Total static normal force of tractor vehicle,
- P_m Pressure at "Brake" coupling head.

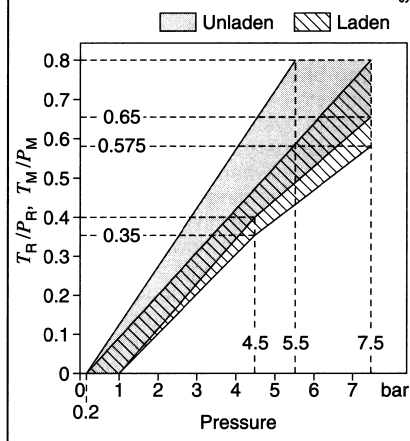
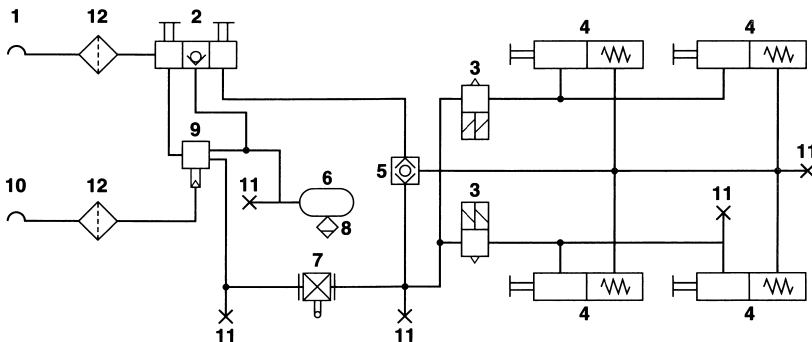


Figure 2: Compressed-air brake system of a 2-axis semitrailer with ABS (simplified representation)

- 1 "Supply" coupling head (red), 2 Double release valve, 3 1-channel ABS pressure-control valve,
- 4 Combination brake cylinder, 5 Shuttle valve, 6 Air reservoir, 7 Load-sensitive braking-force regulator,
- 8 Drain valve, 9 Trailer brake valve, 10 "Brake" coupling head (yellow), 11 Test connection, 12 Line filter.



performed by a standardized electrical plug-in connection in accordance with ISO 7638 [3]; this plug-in connection may have 5 or 7 pins.

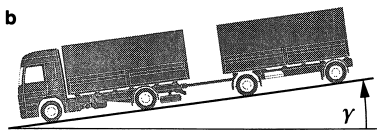
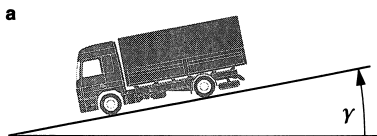
Tractor vehicles and trailers must be freely exchangeable. Compatibility conditions have therefore been defined in Annex 2 RREG 71/320 and ECE R13. Accordingly, the ratio between retardation and pressure at the "Brake" coupling head in the range depicted in Figure 3 must be in the range of 0.2 to 7.5 bar at the "Brake" coupling head. The diagram only applies to tractor vehicle and trailer. All other vehicles and vehicle combinations are covered by other diagrams.

Parking-brake system

The parking-brake system is an independent brake system which must hold the vehicle stationary after it has been brought to a stop, even when the driver is not in the vehicle. The holding-stationary effect is calculated on a downhill gradient with fully laden vehicles. The downhill gradient for Category M, N, O single vehicles (except O1) is 18%. If the vehicles are equipped to tow a trailer, the holding-stationary effect must also be achieved with an unbraked trailer. The downhill gradient is then only 12% (Figure 4).

Figure 4: Test condition for parking-brake system

- a) Single vehicle, 18% downhill gradient.
- b) Tractor vehicle and trailer, 12% downhill gradient; only the tractor vehicle is braked.
- γ Gradient angle.



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The parking-brake system in commercial vehicles and motor buses is usually designed as a spring-loaded brake system. The springs in the spring-type brake cylinders, when the wheel brakes are adjusted in accordance with the regulations, generate the same force as the pneumatic brake cylinders in the service-brake system when the nominal pressure (brake-system calculated pressure) acts on their nominal effective area. If certain malfunctions occur, e.g. brake-circuit failure or energy-source failure, the spring-loaded brakes may not brake automatically and must therefore be protected and designed accordingly.

Energy-assisted parking-brake systems (spring-loaded brake systems) must be equipped with at least one emergency release device. This may be a mechanical, pneumatic or hydraulic device. The parking-brake system must then only be designed for graduated (metered) operation if it has to be engaged to achieve the prescribed secondary braking effect.

In the trailer, the parking-brake system frequently operates as a muscular-energy brake system. If the trailer control system in the tractor vehicle is configured in such a way that the service brake in the trailer also responds when the parking brake is operated in the tractor vehicle (trailer control valve with port 4.3), the parking-brake valve must be provided with a test setting. This makes it possible to release the service brake in the trailer when the parking brake in the tractor vehicle is operated. This in turn makes it possible to check whether only the tractor vehicle braked with the parking-brake system can hold the entire vehicle combination.

Secondary-brake system

There is no independent secondary-brake system. It comes into effect if a malfunction, e.g. brake-circuit failure or energy-source failure, occurs in the service-brake system. In this event, it must still be possible for at least two wheels (not on the same side) to be braked.

The brake system in the trailer too must not be affected by this malfunction. For this reason, the brake systems and activation of the trailer have dual-circuit designs.

The supply volume must be designed such that, in the event of an energy-source failure, after eight full operations of the service brake there is still enough pressure available to achieve the secondary braking effect on the ninth full operation. In the event of a failure of a brake circuit on the supply side, it is essential to ensure that, when the energy source is intact, the pressure in the intact brake circuits does not drop permanently below the nominal pressure. This is achieved by using special protective devices, e.g. a four-circuit protection valve or an electronic unit.

Continuous-operation brake system

The wheel brakes used are not designed for continuous operation. Prolonged braking (e.g. on hill descents) can result in the brakes being thermally overloaded. This causes a reduction in braking effect ("fading") or, in extreme cases, in complete brake-system failure.

A wear-free brake system is referred to as a continuous-operation (retarder) brake system. In Germany, they are required in accordance with StVZO §41 s. 15 [4] for motor buses with a gross weight rating of more than 5.5 t and for other motor vehicles with a gross weight rating of more than 9 t. A retarder must hold a fully laden vehicle over a distance of 6 km and a downhill gradient of 7% at a speed of 30 km/h.

The service brake must be designed accordingly for trailers. Operation of the retarder in the tractor vehicle must not cause the service brake in the trailer to be operated (see also StVZO §72 [5]).

Components of commercial-vehicle brake systems

Air supply and air processing

Air supply and processing comprises the energy source, pressure control, air processing, and compressed-air distribution.

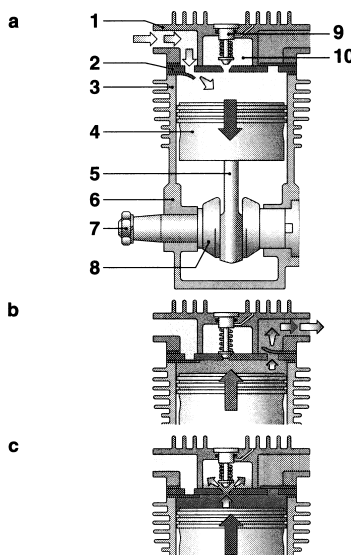
Compressor

A compressor is the source of energy. It takes in air and compresses it to compressed air, the working medium for the brake systems and the ancillaries (e.g. pneumatic suspension, door-closing system).

The compressor is a plunger pump where the crankshaft is driven directly by the vehicle engine (Figure 5). It is fitted to

Figure 5: Compressor

- a) Induction,
b) Compression and discharge,
c) Compression into the additional clearance.
1 Cylinder head, 2 Intermediate plate (with inlet and outlet valves), 3 Cylinder, 4 Piston, 5 Connecting rod, 6 Crankcase, 7 Drive 8 Crankshaft, 9 ESS valve (Energy-Saving System), 10 Additional clearance.



the vehicle engine by means of a flange. Its component parts are:

- The crankcase, which forms a monobloc unit with the cylinder. It contains the crankshaft with connecting rod and piston.
- The cylinder head with intake and pressure connections, as well as connections for water cooling.
- The intermediate plate with inlet and outlet valves.

To reduce losses in the idle mode (opening and flow resistances in valves and lines), an energy-saving system (ESS) is used; this activates a clearance and thus reduces compression work. This reduces fuel consumption.

During its return stroke, the piston draws in air after the inlet valve opens automatically due to vacuum. The inlet valve closes at the start of the piston's return stroke. In the fore stroke, the piston compresses the air. When a certain pressure is reached, the outlet valve opens and compressed air is supplied to the brake system.

Nowadays, compressors achieve a stroke displacement of up to 720 cm³, a pressure

level of up to 12.5 bar, and a maximum speed of 3,000 rpm. Their features include high efficiency, low oil consumption, and a long service life.

Pressure regulator

The pressure regulator controls the compressed air supplied by the compressor in such a way that the operating pressure lies within the activation and cutoff pressure (Figure 6).

As long as the pressure in the compressed-air reservoirs lies below the cutoff pressure, connections 1 and 2 are connected and compressed air passes the pressure regulator. Once the cutoff pressure is reached, the pressure regulator switches to idle mode. Here, the venting piston is activated and connection 1 is connected to the atmosphere (venting).

Air drier

The air drier cleans the compressed air and dries it to prevent corrosion and freezing in the brake system during winter operation.

Basically, an air drier consists of a desiccant box and a housing. The housing incorporates the air passage, a bleeder valve, and a control element for granulate regeneration (Figure 7). The granulate is regenerated by activating a regeneration-air tank.

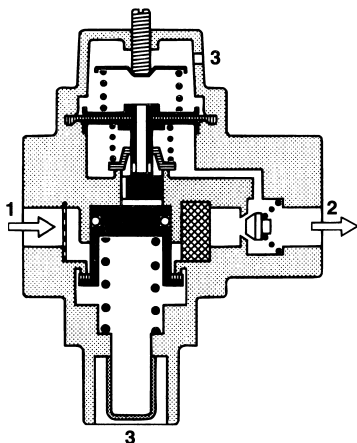
When the bleeder valve is closed, compressed air from the compressor flows through the desiccant box and from there to the supply-air reservoirs. At the same time, a regeneration-air tank is filled with dry compressed air. As compressed air flows through the desiccant box, water is removed by means of condensation and adsorption.

The granulate in the desiccant box has a limited water absorption capacity and must therefore be regenerated at regular intervals. In the reverse process, dry compressed air from the regeneration-air reservoir is reduced to atmospheric pressure via the regeneration throttle upstream of the air drier, flows back through the moist granulate from which it draws off the moisture, and flows as moist air via the open bleeder valve to the atmosphere.

The pressure regulator and air drier can be combined into one unit.

Figure 6: Pressure regulator

- 1 From compressor,
- 2 To the compressed-air reservoirs,
- 3 Venting.



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Four-circuit protection valve

The four-circuit protection valve distributes the compressed air to the various brake and ancillary circuits, isolates the circuits from one another, and ensures the water supply for the remaining circuits in the event of failure of a circuit (Figure 8).

The function of the four-circuit protection valve is provided by overflow valves specially developed for this application. In contrast to a normal overflow valve, the design of this type of overflow valve features two different effective areas on the intake-flow side. The incoming pressure from the pressure regulator acts on one effective area, while the pressure available in the pneu-

matic circuit acts on the other. The opening pressure of the overflow valves is thus dependent on the (residual) pressure of the assigned pneumatic circuit.

The overflow valves can be arranged differently. Often service-brake circuits 1 and 2 and ancillary circuits 3 and 4 are connected in pairs in succession. This ensures that at least one of the two service-brake circuits is filled as a matter of priority.

Figure 7: Air drier with integrated pressure regulator

- 1 Desiccant box, 2 Compression spring, 3 Desiccant, 4 Cup (control valve), 5 Compression spring, 6 Pin, 7 Diaphragm, 8 Compression spring, 9 Heating element, 10 Bleeder valve, 11 Drain connection, 12 Throttle, 13 Non-return valve, 14 Preliminary filter, 15 Secondary filter.

Ports:

- 1 From compressor,
- 21 To air reservoir,
- 22 To regeneration-air tank,
- 3 Vent.

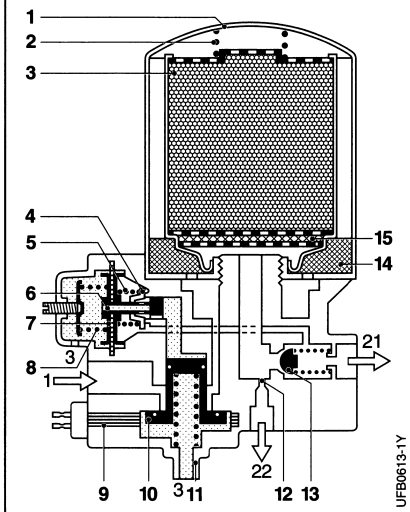


Figure 8: Four-circuit protection valve

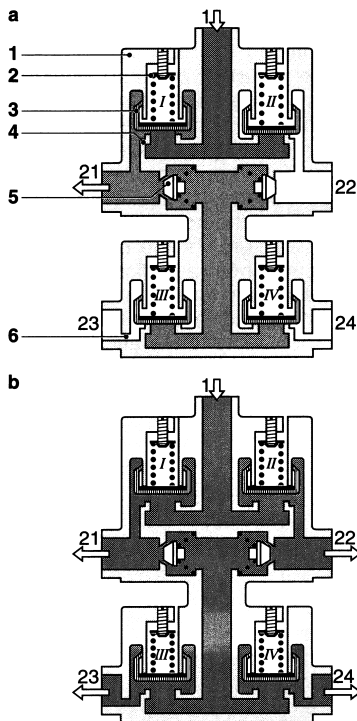
- a) Filling a compressed-air reservoir,
- b) Filling all compressed-air reservoirs.

- 1 Housing,
- 2 Compression spring,
- 3 Diaphragm piston,
- 4 Valve seat,
- 5 Non-return valve,
- 6 Fixed throttle.

I – IV overflow valves

Ports:

- 1 Energy input,
- 21–24 Energy output to circuits 1–4.



The ancillary circuits for this type of valve are additionally protected by two non-return valves. These non-return valves can be omitted from four-circuit protection valves with central intake flow. These overflow valves may also be provided with variable flow restrictors. These enable an empty system to be filled with small quantities of air.

If a malfunction occurs for example in circuit 1 (circuit failure due to a leak), the pressure drops initially only in circuit 1 to 0 bar and in circuit 2 to the closing pressure. The pressure in circuits 3 and 4 is initially maintained by the effect of the non-return valves, but will also drop through consumption to the closing pressure. The intact circuits continue to be supplied under subsequent delivery by the compressor, because the residual pressure in circuits 2, 3 and 4 acts on the secondary effective area of the corresponding overflow valves. The intact circuits are filled again until the opening pressure of the defective circuit (circuit 1) acts on the primary effective area of the corresponding overflow valve, opening this valve. A further pressure increase is not possible, because from this moment the delivered compressed air is lost through the defective circuit. The opening pressure via the primary effective area is adjusted in such a way that it is equal to or above at least the nominal pressure (calculated pressure) of the brake system. This ensures both a sufficient supply of compressed air for the intact service-brake circuit and the secondary braking effect. The supply to the ancillaries such as, for example, trailer, parking-brake system and pneumatic suspension is also maintained.

Electronic air-processing unit

Nowadays, the pressure control, air processing, and compressed-air distribution are combined in one electronic unit, the air-processing unit. The electronic air-processing unit (EAC, Electronic Air Control) is a functional agglomeration of the pressure regulator, air drier and multiple-circuit protection valve into one mechatronic device. In addition, the control system of the parking brake is integrated in part. In all, integrating many functions in a mechatronic unit offers significant benefits with regard to system expenditure, functionality, and energy saving.

Energy storage

The energy required for the braking operation and for the function of the ancillaries is provided and stored in sufficient quantities in compressed-air reservoirs approved for use in road vehicles. The volume must be designed such that, without subsequent delivery, after eight full brakings the secondary braking effect prescribed for this vehicle is still achieved at least by the ninth full braking.

Despite the use of an air drier, the compressed-air reservoirs are equipped with manual or automatically acting drain devices. Compressed-air reservoirs are subject to the requirements of §41a s.8 [4] in conjunction with §72 StVZO [5], and must be approved for use and permanently identified.

The supply systems for the brake systems must be fitted with warning devices. The following requirements apply:

- Red warning light,
- visible to the driver at all times
- comes on no later than on brake application or if the pressure in the supply system for the service brake has dropped to 65 % nominal pressure. 80 % nominal pressure applies to the supply system for the parking-brake system (spring-loaded brake).

Service-brake valve

Service-brake valves (Figure 9) have a dual-circuit design and control the service-brake circuits according to the control force (force-controlled valves).

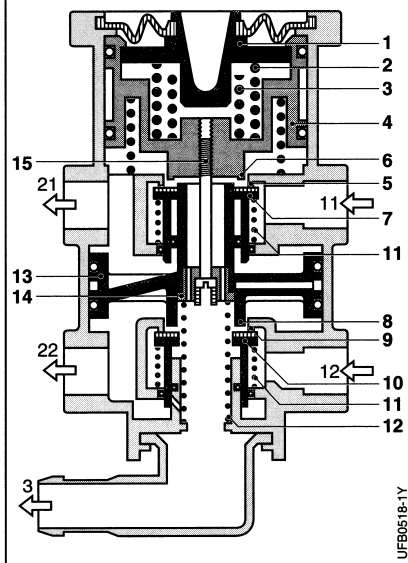
Circuit 1 is actuated by the control device, the push rod and the compression springs (travel-compensating springs). The reaction piston is forced downwards, first closing the outlet valve and then opening the inlet valve. Compressed air is admitted into brake circuit 1 and the pressure increases. The brake pressure acts in the upward direction against the reaction piston, forcing it against the compression springs as long as the partial braking range is not exceeded. The brake end position is reached, with an equilibrium of forces existing at the reaction piston.

Circuit 2 is controlled by the brake pressure in circuit 1. This acts, instead of the control device from above, on the reaction piston of circuit 2. At approximately the same time, the brake end position is reached in circuit 2 as well. In the full-braking position or in the event of a failure of circuit 1, both reaction pistons are mechanically moved to their full extent by means of the control device. The outlet valves are closed, and the inlet valves remain open. Circuits 1 and 2 are pneumatically fully and safely isolated from each other.

Special designs facilitate different controlled brake pressures for circuits 1

Figure 9: Service-brake valve

- 1 Push rod,
 - 2 and 3 Compression springs,
 - 4 Reaction piston,
 - 5 and 9 Inlet-valve seat,
 - 6 and 8 Outlet-valve seat,
 - 7 and 10 Valve plates,
 - 11 Valve springs,
 - 12 Return spring,
 - 13 Control plunger,
 - 14 Spring seat,
 - 15 Connecting rod.
- Ports:
- 3 Vent,
 - 11 Energy input, circuit 1,
 - 12 Energy input, circuit 2,
 - 21 Brake pressure, circuit 1,
 - 22 Brake pressure, circuit 2.



and 2. These are required if a dual-circuit booster cylinder is actuated by the service-brake valve, or if circuit 2 is subject to load-sensitive control. This is made possible by installing an appropriate spring assembly or a reaction piston with several effective areas.

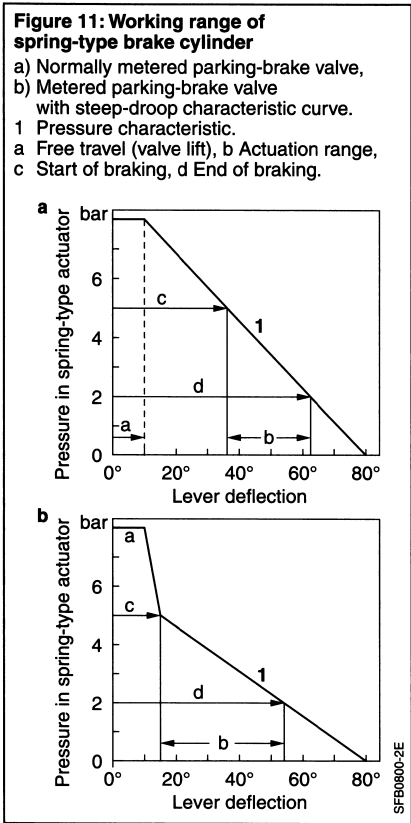
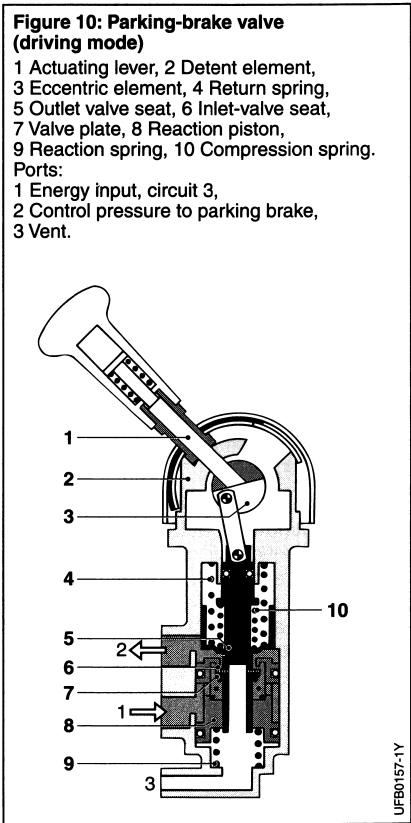
Parking-brake valve

Parking-brake valves (Figure 10) control the pressure in the spring-type brake cylinders as a function of the lever travel (travel-controlled valves). The lever must be permanently and securely lockable in the brake-applied position. Parking-brake valves must only operate under graduated (metered) application when the effect of the parking brake is required to achieve the secondary braking effect. Parking-brake valves must be provided with a test setting when the service-brake system is activated in the trailer on actuation of the parking brake.

Different variants of parking-brake valve exist, depending on their application: non-graduated, graduated or graduated with steep-droop characteristic curve. The latter variant provides for a

highly sensitive graduable effect, because the operating range of spring-type brake cylinders, considered over the lever angle of the parking-brake valve of approximately 80°, is optimally used. The operating range of spring-type brake cylinders is between approximately 5 bar (start of braking) and approximately 2 bar (end of braking), see diagrams in Figure 11).

In pneumatic high-pressure brake systems (operating pressure greater than 10 bar), the parking-brake valve can be fitted with a pressure limiter so that standard spring-type brake cylinders can be used. The facility in parking-brake valves for attaining the capability of metering the controlled pressure is similar to the facility in service-brake valves, but operates in the opposite direction, because the spring-type brake cylinders are ventilated



in driving mode and the brake-applied mode is achieved by bleeding.

Parking-brake valves can have a dual-circuit design. The system is supplied in this case from circuit 3 and – the pneumatic auxiliary release device of the spring-type actuators – from circuit 4. An additionally required rotary-knob, shuttle or check valve can be omitted.

In the version with steep-droop characteristic curve (Figure 11), the start of braking is attained earlier and the actuation range is significantly greater. This is particularly advantageous when the parking brake is used as a secondary brake.

As an alternative to the pneumatic parking-brake valve, there are electronically controlled parking-brake systems (Electronic Parking Brake, EPB). These consist of an EPB module (which can optionally also be integrated in the air-processing unit) and an operating unit. The EPB module contains a bistable valve that can be controlled via integrated solenoid valves and further solenoid valves for executing the trailer test function.

The parking brake can be applied and released using the operating unit, whereby the last active state is retained even after the supply voltage is shut off. Since the operating unit must have a graduable design, the parking brake can also be operated in stages and consequently function as a secondary brake.

As well as manual activation by operating unit, the EPB also provides for a series of comfort and convenience functions, e.g., automatic application when the vehicle is at a standstill (Autopark) and automatic release when starting (Autorelease).

Automatic load-sensitive braking-force regulator

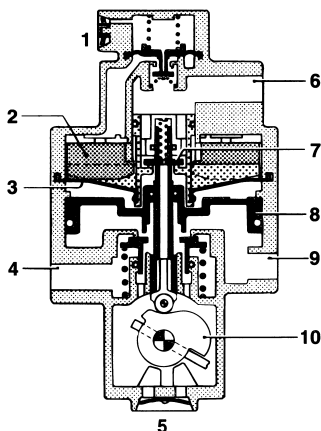
A facility frequently used in the service-brake system of commercial vehicles with pneumatically controlled service-brake systems is automatic load-sensitive braking-force control (ALB). Valves that handle the braking-force distribution enable adjustment of the braking forces to the low axle loads in the partially laden and unladen state and thus a correction of the braking-force distribution on the axles of an individual vehicle or a certain braking level in road trains or semitrailers.

The braking-force regulator (Figure 12) is connected between service-brake valve and brake cylinder. Depending on vehicle payload, it regulates the applied braking pressure. The device has a transfer diaphragm with variable effective area. The diaphragm is held in two radially arranged, interlocking rakes. Depending on the vertical position of the control-valve seat, there is a large reaction area (valve position at bottom) or a small reaction area (valve position at top). Consequently, the brake cylinders are supplied via an integrated relay valve with a reduced pressure which is lower than (unladen), or which is the same as (fully laden) the pressure coming from the service-brake valve. The control-valve can be moved into the load-sensitive position by means of an eccentric element that is connected via linkage to the vehicle axle or by means of a wedge (in the case of vehicles with pneumatic suspension).

The pressure limiter which is integrated into the device at the top allows a small

Figure 12: Braking-force regulator with relay valve

- 1 Vent, 2 Rake, 3 Transfer diaphragm,
- 4 Energy input from air reservoir,
- 5 Vent, 6 Uncontrolled pressure from service-brake valve, 7 Control valve,
- 8 Relay piston, 9 Controlled brake pressure to brake cylinders, 10 Rotary cam.



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partial pressure (approx. 0.5 bar) to flow in to the top of the diaphragm. Thus, up to this pressure there is no reduction in brake-cylinder pressure. This results in the synchronous application of the brakes on all vehicle axles.

As an alternative to the pneumatic ALB valve, pneumatically controlled brake systems are increasingly making use of the EBD function (Electronic Brakeforce Distribution) of ABS, with which braking-force distribution is optimized as a function of the wheel slip (see Wheel-slip control systems).

The electronically controlled brake (EBS) has become the predominant system used in heavy European commercial vehicles. This system performs braking-force distribution electronically as a function of the laden state and other parameters (see Electronically controlled brake system)

Combination brake cylinder

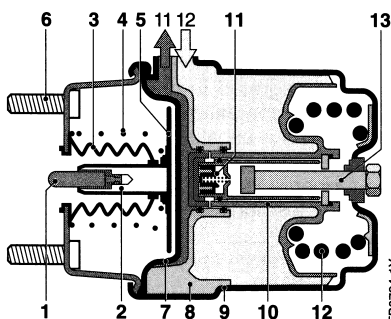
The combined cylinder in the commercial vehicle consists of a diaphragm cylinder part for the service brake and a spring-type actuator part for the parking brake (Figure 13). They are arranged one behind the other and exert force on a joint push rod. A distinction can be made between combined cylinders for S-cam brakes, wedge-actuated brakes, and disk brakes based on the type of wheel brake.

The two cylinders can be actuated independently of one another. Simultaneous actuation results in the addition of their forces. This can be prevented by installing a special relay valve in order to automatically prevent mechanical overloading of other downstream components (e.g. brake drums).

A central release screw allows for a tensioning of the spring of the spring-type brake cylinder without compressed air having to be applied (mechanical emergency release device). This is necessary to assist fitting or, in the event of failure of the compressed air, to be able to maneuver the vehicle.

Figure 13: Combination brake cylinder for disk brake (driving mode)

- 1 Pressure pin, 2 Piston rod,
 - 3 Bellows with seal to disk brake,
 - 4 Compression spring (diaphragm cylinder),
 - 5 Piston (diaphragm cylinder),
 - 6 Housing with fastening bolts,
 - 7 Diaphragm, 8 Intermediate flange,
 - 9 Cylinder housing (spring-type brake actuator),
 - 10 Piston (spring-type brake actuator),
 - 11 Bleeder valve (spring-type brake actuator chamber),
 - 12 Compression spring (spring-type brake actuator),
 - 13 Release device (spring-type brake cylinder).
- Air ports: 11 Service brake, 12 Parking brake.



When the service brakes are operated, compressed air flows into the diaphragm cylinder and presses the plunger disk and the push rod against the lever in the disk brake. A drop in air pressure releases the brake.

When compressed air flows into the spring-type actuator part, the piston presses the springs together and the brake is released. If the chamber is vented, the spring-type brake cylinder exerts a force via the piston rod on the diaphragm part and presses the push rod into the mechanism of the disk brake via the piston disk.

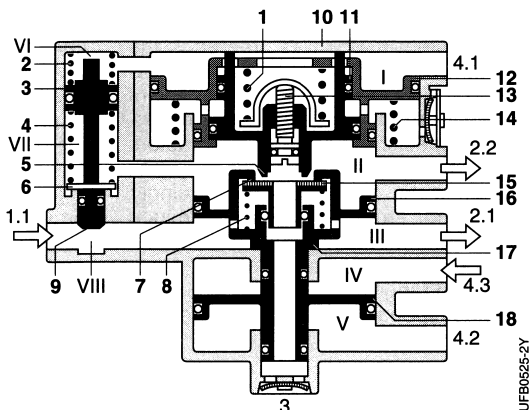
Figure 14: Trailer control valve with decoupling function (driving mode)

- 1 and 2 Compression spring, 3 Control plunger, 4 Spring assembly, 5 Outlet-valve seat, 6 Disk, 7 Inlet-valve seat, 8 Compression spring, 9 Throttle pin, 10 Housing, 11 and 12 Control plunger, 13 Adjusting screw, 14 Compression spring, 15 Valve disk, 16 Reaction piston, 17 Collar, 18 Control plunger.

I – VIII Chambers.

Ports:

- 1.1 Energy input from circuit 3,
2.1 Energy output to "Supply"
coupling head (red),
2.2 Energy output to "Brake"
coupling head (yellow),
4.1 Control port
uncontrolled pressure circuit 1,
4.2 Control port
uncontrolled pressure circuit 2,
4.3 Control port parking brake,
3 Central vent.



Trailer control valve

The trailer control valve installed in the tractor controls the trailer's service brake. This multi-circuit relay valve is triggered by both service-brake circuits and by the parking brake (Figure 14). In the driving mode, supply chamber III and chamber IV of the parking-brake circuit are under the same pressure. The brake line to the trailer is connected to the atmosphere via the central venting. A pressure increase in chamber I of brake circuit 1 and in chamber V of brake circuit 2 leads to the corresponding pressure increase in chamber II for the brake line to the trailer. A pressure drop in both brake circuits also leads to

the same pressure drop in the brake line. Operation of the parking-brake system leads to venting of the parking-brake circuit (chamber IV). This increases the pressure in chamber II for the brake line to the trailer. When air enters chamber IV, the brake line is vented again.

If the brake line to the trailer is pulled off, it is prescribed that the pressure in the supply line to the trailer must have fallen to a pressure of 1.5 bar in less than two seconds (RREG 71/320). To achieve this, the compressed-air supply to the supply line is throttled by means of an integrated valve.

Electronically controlled brake system

Requirements and functions

With the further development of the dual-line compressed-air brake system, an electronic (or electronically controlled) brake system (EBS) was created in the mid-1990s. On account of the modular design, it is possible to cover different vehicle types with just a few components. Vehicle-specific differences and factors can be covered to a large extent by programming the central ECU accordingly. The control arrangement is determined by the number of axles and their arrangement and the required scope of functions, and ranges from 4S/4M to 8S/6M (S wheel-speed sensor, M pressure-control module).

Design and operating principle

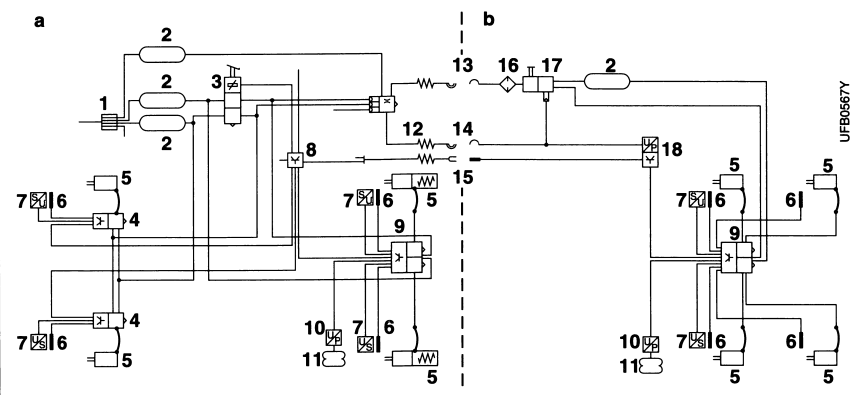
The electronic brake system (Figure 15), like a conventional compressed-air brake with antilock braking system (ABS), comprises a compressed-air supply system, but the pressure-regulator, air-drier and multiple-circuit protection-valve functions can be combined in an electronic unit (EAC, Electronic Air Control). It is thus possible to adapt certain functions – such as e.g. fill sequences or regeneration – better to the required conditions and to guarantee even greater functional reliability.

In the electronic brake system too, energy is stored in compressed-air reservoirs and from there made available to the pressure-control modules and to the service-brake valve. The service-brake valve consists of an electric pedal-travel sensor and a pneumatic section which is functionally identical to the previous design. The pedal-travel sensor consists of a redundant travel sensor (e.g. two redundantly arranged electrical potentiometers or a redundant inductive travel sensor) which is deflected via the control device and delivers the redundant output signal to the central ECU. From this the ECU cal-

Figure 15: Service-brake system of an electronically controlled braking system

a) Tractor vehicle, b) Trailer.

- 1 Four-circuit protection valve, 2 Air reservoir, 3 Service-brake valve with braking-level sensor,
- 4 Single-channel pressure-control module, 5 Brake cylinder, 6 Wheel-speed sensor,
- 7 Brake-lining wear sensor, 8 EBS ECU in tractor vehicle,
- 9 Two-channel pressure-control module, 10 Pressure sensor, 11 Air-spring bellows,
- 12 Trailer control valve, 13 "Supply" coupling head (red),
- 14 "Brake" coupling head (yellow), 15 ISO 7638 plug-in connection (7-pin), 16 Line filter,
- 17 Trailer brake valve with release device, 18 EBS ECU in trailer.



culates for each wheel an individual brake pressure and in turn activates the pressure-control modules on the individual axles so that the required brake pressure is applied to the brake cylinders downstream of the pressure-control modules. The applied brake pressure is regulated with the aid of integrated pressure sensors in the pressure-control modules. A pneumatic brake-pressure generation is performed in parallel in the pneumatic part of the service-brake valve which on the one hand determines the brake feel and on the other hand acts as a fallback level in the event of an electrical fault.

The brake-pressure modules are available as one-channel or two-channel designs. If the vehicle is set up to tow a trailer, a trailer control module is also provided as a substitute for the trailer control valve. This trailer control module is likewise activated in the braking operation by the central ECU and makes an adapted control pressure available at the "Brake" coupling head (yellow). This makes it possible to carry a conventionally braked trailer. If a trailer is carried with an independent electronic brake system, this is controlled by an electrical connection via the plug-in connection as per ISO 7638 (ABS connector). The trailer must nevertheless also be pneumatically coupled, because this is the only way that the trailer can be supplied with pressure and in the event of a system failure pneumatically controlled. By controlling the electronic brake system in the trailer, it is possible to provide optimum matching with regard to braking performance between the tractor vehicle and the trailer. Simultaneous and matched braking performance facilitate optimized coupling-forcing matching.

Further functions, such as antilock braking system (ABS), traction control system (TCS) and driving-dynamics control system (Electronic Stability Program),

are integrated into the electronic brake system's scope of functions. The turning behavior of the wheels is monitored by the wheel-speed sensors and the antilock braking system. Depending on the design, the information is made available to the central ECU or to the pressure-control module, where it is processed. In the event of incipient wheel locking, depending on the system arrangement and design, a control intervention is effected via the pressure-control modules or by downstream pressure-control valves in accordance with the control variants known to the ABS system (individual control, modified individual control, or select-low control). Intervention by the traction control system when the wheels spin takes the form of an engine and brake intervention. Further sensors are needed for the functions of the driving-dynamics control system. The steering-wheel angle is recorded by a steering-angle sensor. A yaw-velocity sensor, also known simply as a yaw sensor, records the rotational speed about the vehicle vertical axis. A lateral-acceleration sensor also records the lateral acceleration. When the data have been evaluated, swerving or jackknifing is detected and stabilized by the specific introduction of brake pressure into the relevant brake cylinders and intervention in other systems (see Driving-dynamics control for commercial vehicles).

If an electrical fault occurs, the vehicle can be braked by means of one or two redundant pneumatic circuits with at least the demanded secondary braking effect, and the trailer brake system controlled.

Optimum cooperation between all the systems can be achieved through data communication with other systems in the vehicle and the trailer. Optimized deceleration and acceleration processes and additional functions can be realized in this way.

Advantages of electronic brake systems in a commercial vehicle are:

- Fast and simultaneous brake-pressure buildup in all the brake cylinders,
- good metering capability, thus optimum braking comfort,
- optimum matching between tractor and trailer through control of coupling forces,
- exact braking-force distribution,
- uniform brake-lining wear,
- ABS, TCS and ESC functions are integrated (brake and engine interventions), traction control for offroad applications can be easily realized,
- driving-dynamics control via engine and brake interventions when oversteering or understeering is detected, in response to the risk of swerving (articulated road train, articulated bus), and intervention in response to the risk of overturning,
- ease of servicing thanks to extensive diagnosis functions.

Components of the electronic brake system

Electronic control unit (ECU)

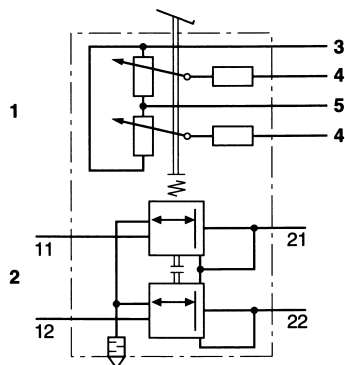
The control point of an electronic brake system consists of a central ECU in which all the system functions run. Aside from the cab-mounted variants, frame-mounted ECUs are increasingly gaining acceptance. The benefit of the latter is that the sensors for yaw rate and lateral acceleration can be integrated and consequently there is no need for a separate sensor to be fitted.

Service-brake valve

A service-brake valve for the electronic brake system is similar in design to conventional, purely pneumatic service-brake valves. However, in the service-brake valve the electronic setpoint values for brake-pressure control are also recorded (Figure 16). It thus fulfills two functions: A redundant sensor detects the driver's brake request by measuring the operating travel of the valve tappet. The measured value is transmitted to the central ECU and converted there into a braking request. In the same way as a conventional service-brake valve, the pneumatic control pressure is applied according to the actuation travel. These control pressures are required for "backup" control in the event of a fault.

Figure 16: Service-brake valve with two pneumatic control circuits

- 1 Braking-level sensor,
 - 2 Service-brake valve,
 - 3 Supply connection,
 - 4 Electrical potentiometer connections,
 - 5 Electrical ground connection.
- Pneumatic ports:
 11 Energy input, circuit 1,
 12 Energy input, circuit 2,
 21 Back-up control pressure, circuit 1,
 22 Back-up control pressure, circuit 2.



Pressure-control modules

The pressure-control modules (Electro-Pneumatic Modulator, EPM, Figure 17) are the interface between the electronic brake system and the pneumatically actuated wheel brakes. They convert the required braking pressures transmitted via the CAN bus into pneumatic pressures. This conversion operation is usually performed with an inlet and outlet solenoid combination. A pressure sensor measures the controlled brake pressure and thus facilitates brake-pressure control in the closed control loop. The electrically activated "backup" valve shuts off the pneumatic control pressures of the service-brake valve in order to permit interference-free electrical pressure control.

Figure 17: Single-channel pressure-control module

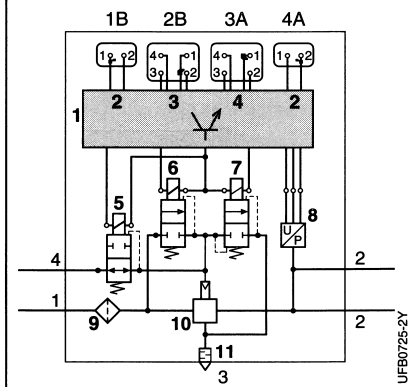
- 1 ECU, 2 Speed sensor,
- 3 Brake-lining sensor,
- 4 CAN, 5 "Backup" valve,
- 6 Inlet valve, 7 Outlet valve,
- 8 Pressure sensor, 9 Filter,
- 10 Relay valve, 11 Muffler.

Connectors:

- 1B Connector, speed sensor 1,
- 2B Connector, brake-lining sensor,
- 3A Connector, supply and CAN bus,
- 4A Connector, speed sensor 2.

Ports:

- 1 Energy input,
- 2 Brake pressure to brake cylinder,
- 3 Vent,
- 4 Back-up control input.



Mounting the pressure-control modules close to the wheels means that the electrical wires for connecting the wheel-speed sensors and the brake-lining wear sensors can be kept short. The information is prepared in the pressure-control module and transmitted via the CAN bus to the central ECU. This reduces the outlay on cabling in the vehicle.

Trailer control module

The electronic trailer control module (Trailer Control Module, TCM) enables modulation of the trailer control pressure according to the functional requirements of the electronic brake system. The limits of the electrical control ranges are defined by legal requirements. The electronically specified setpoint value is converted into a physical braking pressure by means of a solenoid arrangement similar to that in the pressure-control module. The "backup" pressure is shut off either by a "backup" solenoid or by pneumatic retention, depending on the type of design adopted.

Under all normal conditions, the trailer control module must be activated by two independent control signals. This may be two pneumatic signals from two control circuits, or one pneumatic and one electrical control signal.

References for Brake systems for commercial vehicles

- [1] Directive 71/320/EEC: Council Directive of 26 July 1971 on the approximation of the laws of the Member States relating to the brake systems of certain categories of motor vehicles and their trailers.
- [2] ECE-R13: Uniform provisions concerning the approval of category M, N and O vehicles with regard to brakes.
- [3] ISO 7638: Road vehicles – Connectors for the electrical connection of towing and towed vehicles.
- [4] StVZO (road traffic licensing regulations, Germany) §41: Bremsen und Unterlegkeile (Brakes and wheel chocks).
- [5] StVZO §72: Übergangsbestimmungen (Transition provisions).
- [6] E. Hoepke, S. Breuer (Editors): Nutzfahrzeugtechnik. 8th Ed., Verlag Springer Vieweg, 2016.

Continuous-operation brake systems

Commercial vehicles essentially use two types of continuous-operation brake systems, deployed separately or in combination: the exhaust-brake system and the retarder.

Exhaust-brake systems

The resistance which an engine brings to the speed imposed from the outside without a fuel supply is termed the engine or exhaust brake, or drag power. The drag power of standard engines is 5 to 7 kW per liter displacement. With a pure exhaust brake the regulation according to §41 or the road traffic licensing regulations in Germany (StVZO), s. 15 cannot be observed. Further measures can be used to increase the effect of the exhaust brake.

Exhaust-brake system with exhaust flap

In the exhaust brake with exhaust flap, a valve with a flap closes the exhaust train. The fuel supply is interrupted at the same time. As a result, back pressure is generated in the exhaust-gas system and must be overcome by each piston during its exhaust stroke (Figure 18). The braking

power can be regulated by means of a pressure-control valve in the exhaust train. Additionally, this valve ensures that at high revs an excessively high pressure does not result in valve or valve-gear damage.

The exhaust brake is the most common variant used in trucks and buses, delivering a braking power of 14 to 20 kW per liter displacement.

Exhaust-brake system with constant throttle

The exhaust brake with constant throttle is also known as a decompression brake. In this system, the work performed by the engine in the compression phase is not utilized. The exhaust valves or an additional valve (constant throttle, Figure 19) is/are specifically opened at the end of the compression cycle, thereby relieving the pressure built up in the compression phase. Thus, no further work can be delivered to the crankshaft in the expansion phase.

Engine brake system with exhaust flap and constant throttle

Braking power can be further improved by a combination of exhaust flap and constant throttle (Figure 19). This combination can deliver braking power of 30 to 40 kW per liter displacement.

Figure 18: Exhaust brake with exhaust flap and additional pressure-control valve

- 1 Exhaust-flap actuation (compressed air),
- 2 Exhaust flap,
- 3 Bypass,
- 4 Pressure-control valve,
- 5 Exhaust,
- 6 Intake,
- 7 Piston (4th power stroke, exhaust cycle).

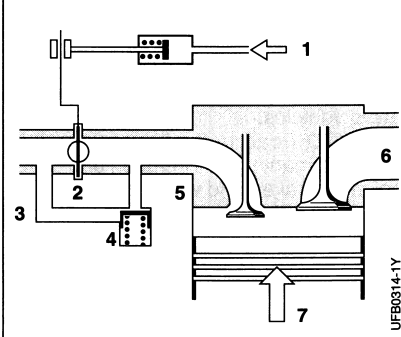
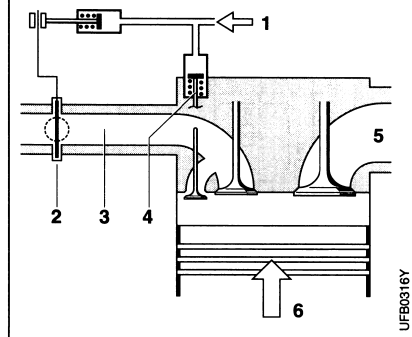


Figure 19: Exhaust brake with exhaust flap and constant throttle

- 1 Compressed air,
- 2 Exhaust flap,
- 3 Exhaust,
- 4 Constant throttle,
- 5 Intake,
- 6 Piston (2nd power stroke, compression cycle).



Retarder

Retarders are wear-free continuous-operation brakes. There are two types that differ in how they operate: hydrodynamic and electrodynamic retarders. Both systems, and the exhaust-brake systems, relieve the load on the service brake system and thereby increase the economic efficiency of the vehicle. The use of a hydrodynamic retarder can increase the service life of the service brake by a factor of 4 to 5.

In modern vehicles, retarders are incorporated into the brake-management system. The exhaust brake and the retarder are often combined as a continuous-operation brake in a vehicle. The brakes must then be activated by means of the electronic brake-management system.

Hydrodynamic retarder

Hydrodynamic retarders, also known simply as hydraulic retarders, can be subdivided into the categories of primary retarders and secondary retarders.

The primary retarder is located between the engine and the transmission, the secondary retarder between the transmission and the powered axle. Both primary and secondary retarders operate in the same way. When the retarder is activated, oil is pumped into the working area. The driven rotor accelerates this oil and transfers it at the outside diameter to the stator. The oil strikes the static stator blades and is decelerated. The oil flows at the inside diameter to the rotor. The rotor's rotary motion is inhibited, and the vehicle is decelerated.

The kinetic energy is primarily converted into heat. For this reason, some of the oil must be permanently cooled by a heat exchanger.

The braking torque can be input using a hand lever or the brake pedal (in the case of a retarder integrated into the EBS electronically controlled brake system). A retarder's braking torque is dependent on the degree of fill in the working area between the rotor and the stator. The degree of fill is regulated by an ECU via a control pressure which is adjusted by proportioning valves.

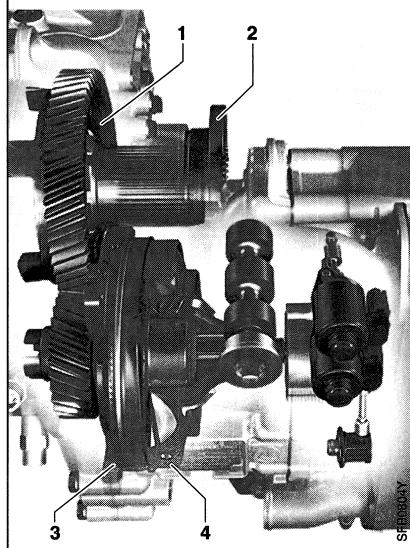
A retarder can be activated hydraulically or pneumatically, whereby the braking torque can be achieved in discrete braking stages and also steplessly. Oil is primarily used as the working medium in a retarder. Current hydraulic retarders can deliver braking power up to 600 kW for brief periods. However, the continuous braking power of a retarder is dependent on the cooling capacity of the vehicle's cooling system. Modern vehicles can dissipate a continuous braking power from a retarder of 300 to 350 kW via the cooling system. Sensors are used to record overheating of the retarder or the cooling system, and if necessary the braking power is reduced under controlled conditions until the braking power equals the dissipatable quantity of heat.

Primary retarder

In the case of a primary retarder located in the drivetrain between the engine and transmission after the converter, force is

Figure 20: Functioning principle of a retarder as illustrated by a ZF interarder by way of example

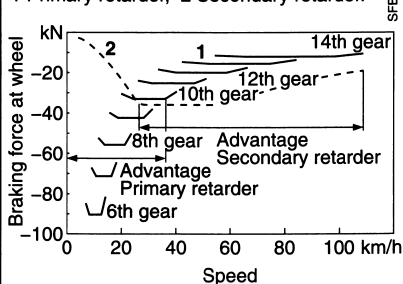
- 1 High-speed stage,
- 2 Output flange,
- 3 Stator, 4 Rotor.



transmitted through the powered axles and the transmission in such a way that the total overrun torque is directed through the transmission. The braking effect of the primary retarder is dependent on the engine speed and the selected gear, but is not dependent on the vehicle's output speed and driving speed. This lack of dependence on the output speed is one of the major advantages of primary retarders. These are highly effective at speeds below 25 to 30 km/h (Figure 21). This is the reason why primary retarders are primarily used in vehicles which are driven at lower average speeds such as city buses

Figure 21: Operating ranges of primary and secondary retarders

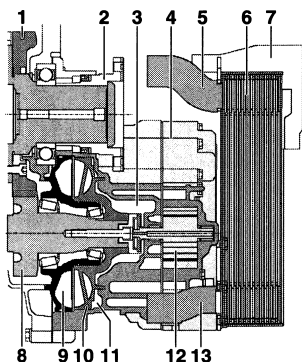
1 Primary retarder, 2 Secondary retarder.



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Figure 22: Hydrodynamic secondary retarder up to 600 kW braking power (ZF)

- 1 High-speed gear, 2 Output flange,
- 3 Intake passage, 4 Control housing,
- 5 Coolant inlet, 6 Heat exchanger,
- 7 Electronics, 8 Pinion shaft, 9 Stator,
- 10 Rotor, 11 Discharge passage,
- 12 Pump, 13 Coolant outlet.



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and municipal vehicles. Their compact design is another advantage. A disadvantage of primary retarders is that the braking force is interrupted during a gear change. The braking force must be reduced during gear changing.

Secondary retarder

In the case of a secondary retarder (Figure 22), which is located after the engine, clutch and transmission, force is transmitted directly via the output shaft. Unlike a primary retarder, there is no interruption of the braking force during a gear change with a secondary retarder. The braking effect is dependent on the ratio of the output shaft and on the driving speed. It is not dependent on the selected gear. The braking torque of a secondary retarder is very much dependent on the rotor speed. For this reason, the rotor speed is often increased by means of a high-speed stage.

The secondary retarder demonstrates great efficiency at speeds over 40 km/h (Figure 21); at speeds below 30 km/h the braking torque falls off dramatically. Because of its design, the secondary retarder can also be subsequently adapted to a transmission. The extra weight of a secondary retarder with accompanying heat exchanger and oil fill is often cited as a disadvantage, since the additional mass reduces the vehicle payload.

Secondary retarders are primarily used in long-distance vehicles which are driven at high average speeds, such as trucks and tour buses.

Electrodynamic retarder (eddy-current brake)

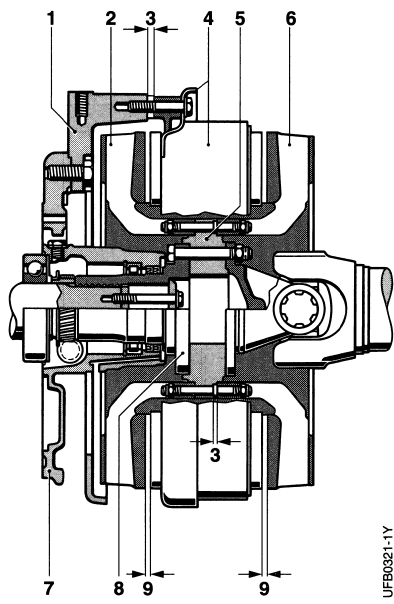
The electrodynamic retarder (Figure 23) contains two non-magnetizable steel disks (rotors) which are connected with torsional strength to the input and output shafts (here: propshaft), and a stator which is equipped with 8 or 16 coils and fastened by means of a star-shaped bracket to the vehicle frame. As soon as electric current (from the alternator or battery) flows through the coils, magnetic fields are generated which are closed by the rotors. These magnetic fields induce eddy currents in the rotors as they turn. These eddy currents in turn generate magnetic fields in the rotors which counteract the exciting

magnetic fields and thus build up a braking effect. The braking torque is determined by the strength of the excitation field, the rotational speed, and the air gap between the stator and the rotors. The braking torque decreases as the air gap increases; this air gap can be adjusted by means of spacers. Shift stages with different braking torques (Figure 24) are obtained by interconnecting the field coils in different configurations. The heat generated is dissipated by convection and radiation to atmosphere via the internally ventilated rotor disks.

As the rotors are increasingly heated, the braking power of the electrodynamic retarder decreases significantly (Figure 25). The retarder's braking power is reduced by thermal protection in order to prevent the retarder from being destroyed by excessive temperature during braking operation.

Figure 23: Electrodynamic retarder

- 1 Star-shaped bracket,
- 2 Rotor, transmission side,
- 3 Spacers (for adjusting air gap),
- 4 Stator with coils, 5 Intermediate flange,
- 6 Rotor, rear-axle side, 7 Transmission cover,
- 8 Transmission output shafts, 9 Clearance gap.



Like the primary retarder, the electrodynamic retarder is distinguished by high braking power at low engine speeds and relative design simplicity. On the downside, however, it can weigh up to 350 kg, depending on its size.

References for continuous-operation brake systems

- [1] E. Hoepke, S. Breuer (Editors): *Nutzfahrzeugtechnik*, 8th Edition, Verlag Springer Vieweg, 2016.

Figure 24: Braking-torque characteristic of an electrodynamic retarder

4a Braking power when the cooling power limit has been reached (switching stage 4).

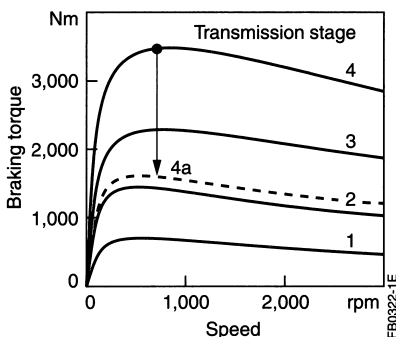
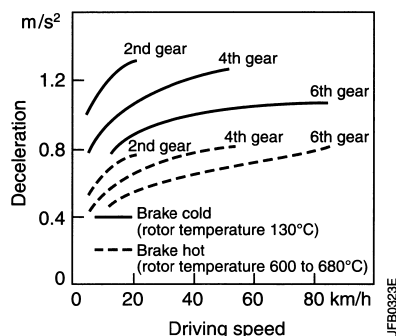


Figure 25: Influence of transmission ratio and rotor temperature on the performance of electrodynamic retarders

17 t commercial vehicle, laden.



Wheel brakes

Wheel brakes are friction brakes that convert kinetic energy into heat energy during braking. Disk and drum brakes are used as wheel brakes. Hydraulic pressure (for passenger cars) and pneumatic pressure and spring force (spring-loaded brake, for heavy commercial vehicles) are converted into an application force to press the brake pads and linings against the brake disks and drums respectively.

In passenger-car applications the thermal demands placed on wheel brakes can, in view of ever-increasing vehicle weights and higher attainable driving speeds, only be satisfied by disk brakes. Drum brakes are now only used in sub-compact-size cars on the rear axle.

In the commercial-vehicle sector disk brakes have gained acceptance for on-road applications in Europe and increasingly in North America as well. In markets with less well developed road infrastructures and in applications with greater off-road usage drum brakes still play a significant role since they react more robustly to dirt and are easier to handle (maintenance, repair, etc.).

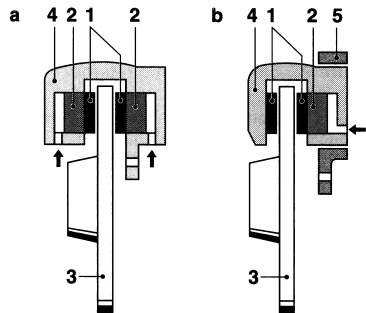
Disk brakes for passenger cars

Functioning principle

Disk brakes generate the braking forces on the surface of a brake disk that rotates with the wheel (Figure 2). The U-shaped brake caliper with the brake pads is mounted to non-rotating vehicle compo-

Figure 2: Disk brakes (diagram)

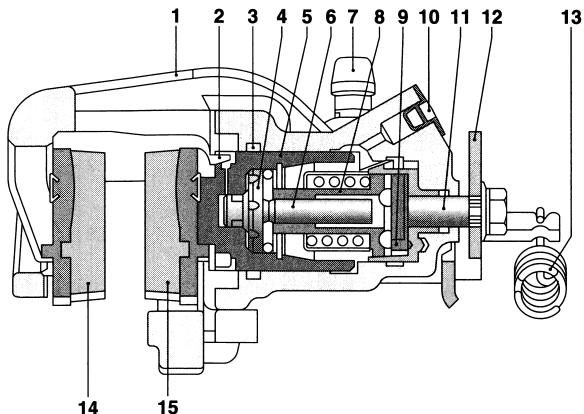
- a) Fixed-caliper brake,
- b) Floating-caliper brake.
- 1 Brake pads,
- 2 Piston,
- 3 Brake disk,
- 4 Brake-caliper housing,
- 5 Brake anchor plate.



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Figure 1: Floating-caliper brake with parking-brake mechanism

- 1 Brake-caliper housing,
- 2 Dust-protection seal,
- 3 Sealing ring,
- 4 Coupling,
- 5 Piston,
- 6 Threaded spindle,
- 7 Bleeder valve,
- 8 Parking-brake mechanism,
- 9 Cam plate,
- 10 Hydraulic port,
- 11 Shaft,
- 12 Parking-brake lever,
- 13 Spring,
- 14 Outer brake pad,
- 15 Piston-side brake pad.



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nents (wheel carrier). The floating-caliper design with or without parking-brake mechanism has proven successful.

Principle of the fixed-caliper brake

In a fixed-caliper brake, both halves of the housing (flange and cover parts) are joined by the housing connecting bolts. Each half of the housing contains a piston to press the brake pad against the brake disk (Figure 2a). Ports in the housing halves connect the two pistons hydraulically.

Principle of the floating-caliper brake

In a floating-caliper brake, a piston presses the piston-side (inner) brake pad against the brake disk (Figure 2b). The generated reaction force moves the brake-caliper housing and thereby presses the outer brake pad indirectly against the brake disk. In this brake caliper the piston is therefore only seated on the inner side.

Floating-caliper brake for passenger cars

The brake caliper can be moved axially in the brake anchor plate and is guided by two sealed guide pins in the brake anchor plate (Figure 1).

Braking with the service brake

The hydraulic pressure generated by the brake master cylinder enters the cylinder chamber behind the piston via the hydraulic connection. The piston is shifted forwards and the brake pad on the piston side is applied to the brake disk. The reaction force that arises shifts the brake-caliper housing mounted in a bolt guide against the direction of piston movement. This also means that the outer brake pad is applied to the brake disk. The path of the brake pads and of the piston covered up to that point is referred to as clearance. Another increase in pressure increases the downforce of the brake pads.

Releasing the service brake

When the piston moves through the clearance, the sealing ring, which is rectangle in its initial position, is deformed. The deformed sealing ring pulls the piston back by the clearance when the hydraulic pressure drops (roll-back effect).

Braking with the integrated parking brake

When the integrated parking brake is operated, the force is transferred via the handbrake cable to the parking-brake lever. This is then twisted and the rotary motion is transferred via the shaft to the cam plate. As the balls run onto the cam plate, the piston is shifted via the pressure sleeve in the parking-brake mechanism; the threaded spindle bolted in this mechanism is shifted towards the brake pad. After crossing the clearance, first the brake pad on the piston side and then the outer brake pad are pressed against the brake disk.

Releasing the parking brake

After releasing the handbrake lever, the parking-brake lever, the shaft and the cam plate turn back to their initial positions. The pressure sleeve, the threaded spindle and the piston are pressed back into their initial position by the springs in the parking-brake mechanism. The final clearance is reached as the sealing ring reassumes its shape.

Automatic self-adjusting mechanism

Wear on the brake pads and brake disks increases the clearance and thus has to be balanced out. This automatic clearance compensation takes place during braking. The inside diameter of the rectangular piston sealing ring is slightly smaller than the piston diameter. The sealing ring thus surrounds the piston with a pre-tension. During braking the piston moves towards the brake disk and tensions the sealing ring, which as a result of its static friction can then slip on the piston only when the piston travel between brake pad and brake disk has in response to abrasion on the brake pads has become greater than the envisaged clearance. When the brake is released the piston is pulled back only by the envisaged clearance. In this way, stepless readjustment to a constant clearance is possible.

The clearance compensation of the parking-brake mechanism also takes place on application of the service brake.

The clearance of a brake caliper is approx. 0.15 mm and is thus in the range of the maximum permissible static disk run-out (axial movement per brake-disk

rotation on account of manufacturing tolerances or bearing clearances).

Brake disks

The energy converted during braking into heat is mainly absorbed by the brake disk and then dissipated to the ambient air. A fundamental distinction is made between solid disks and ventilated brake disks (Figure 3).

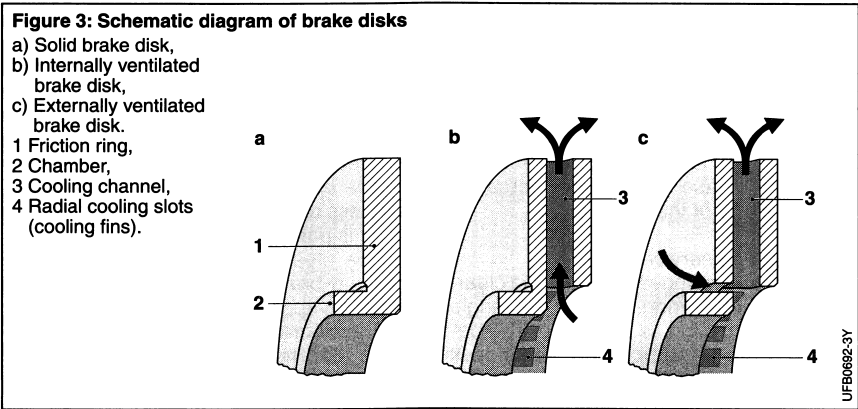
There are two different types of ventilated brake disk: Internally and externally ventilated. On account of the higher thermal load ventilated brake disks are used on the front axle and in high-performance or heavy vehicles also on the rear axle. Perforated or grooved friction rings improve the cooling effect and also improve wet starting response. Ventilated brake disks with radial cooling slots are used to further improve heat dissipation (Figure 3).

Brake disks are usually manufactured from gray cast iron. A higher proportion of carbon improves the heat-absorption capability and noise-damping performance. Alloys containing chrome or molybdenum for example also increase wear resistance. More thermally resistant brake disks made of ceramic containing silicon carbide reinforced with carbon fiber are also used on sports cars and luxury-class cars.

High corrosion requirements are imposed on brake disks due to their installation position directly on the wheel and the high mileage. Brake disks are partially or

completely coated to improve corrosion resistance. High-heat-resisting paints or for example coatings containing zinc are used.

The last few years have also seen the introduction of built-up brake disks. These brake disks have a separation between the friction ring and the chamber area. The chamber area is usually made of a different material (aluminum or sheet steel) in order to achieve weight savings. These brake disks furthermore offer benefits with regard to warping (friction-ring deformation under thermal influence). A distinction is made between screwed, cast, riveted, and combined positive and non-positive connections.



Brake pads for disk brakes**Function and requirements**

The decelerating force during the braking operation is generated by sliding friction between the brake pad and the brake disk. The coefficient of friction denotes the ratio between the tensioning force generated by the brake caliper and the resulting decelerating frictional force between the pad and the disk. For passenger cars it ranges between 0.3 and 0.5 and is included in the chassis-system configuration.

As part of a safety system this function must be performed with absolute reliability and ensue in a predictable way as transparently as possible for the vehicle driver. This therefore requires that the coefficient of friction of the brake pad against the brake disk used always be identical wherever possible under all the varying – even extreme – operating conditions in wide ranges. Furthermore, the wear on the brake pads and the brake disk must be appropriately low. The dust emissions associated with wear should also be kept as low as possible and the composition of these emissions should pollute the environment as little as possible. Brake systems should also operate with as little noise as possible.

Optimum tuning of the brake system to the vehicle is quite significantly achieved via the properties of the brake pad, the design and manufacture of which, of all the components in the system, opens up the most wide-ranging possibilities.

Design

The brake pad (Figure 4) consists of a number of parts whose properties must all be meticulously matched to each other to meet the complex functional requirements. A back plate (carrier) usually made of steel serves to support the braking force exerted by the friction lining against the brake caliper and conversely to transfer the piston force as uniformly as possible to the lining friction surface. To this end, the back plate must adhere to exact dimensional tolerances so as to avoid braking noises or unreliable functioning. In addition, the material must exhibit sufficient strength so as not to permit any plastic deformation at the contact faces or hooks. In many a design the

back plate still supports the strength of the brake caliper.

For the most part, anti-noise shims or films are glued, riveted or even attached with clips to the back of the brake pad painted for corrosion-protection purposes. They are for the most part essential for avoiding noise emissions. It is important here to ensure that the shims cannot be appreciably displaced over the period of operation. Otherwise, unwanted contact between the anti-noise shim and brake or wheel components may result under adverse circumstances. Clips for positioning and locating in the piston and caliper may still be required for fitting in the brake.

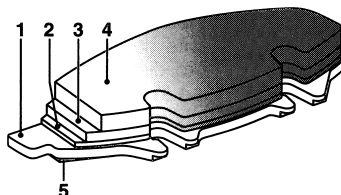
Manufacture

The most cost-effective solution is, after cleaning off oxide coatings and oil residues, to apply a coating of a highly thermally and chemically stable adhesive. Racing cars and a number of heavy, high-performance premium vehicles also use bonding variants which establish positive locking between the pad and the carrier, and thus are above all much more thermally resistant but at the same time significantly more expensive than an adhesive bonding.

Finally, the friction lining is applied. The friction lining is for the most part pressed from a molding powder or granulate in a heated press onto the back plate. At temperatures between 130 °C and around 170 °C the binder component of the friction-lining mixture combines permanently

Figure 4: Structure of a brake pad for disk brakes

- 1 Back plate,
- 2 Adhesive layer,
- 3 Intermediate layer,
- 4 Friction lining,
- 5 Anti-noise shim.



with the back-plate adhesive to form a component. In the process all the other ingredients are also integrated into the binder matrix usually consisting of highly chemically and thermally resistant phenolic resin. This so-called hot-pressing process with dry mixtures has become established in the mass market as the best combination of reliability, range of adjustments of physical properties, product variance, and costs.

Composition of brake pads

The recipe for the molding compound of a typical European brake pad consists of up to more than twenty different ingredients. These can be grouped into the categories of binders, metals, graphite and coke, fillers, organic fibers, abrasives, and lubricants. Table 1 shows by way of example some of the raw materials normally used in brake pads. Aside from their concentration in the recipe, their effect in the brake pad is also determined by their individual composition, microstructure, and particle size. Reliable braking performance with low wear is obtained only by choosing ideal ingredients and carefully optimizing the concentrations of the metal, abrasive, and lubricant proportions. Physical-mechanical properties for sufficient strength and particularly also for comfortable, noise-free braking are adjusted by means of tribologically neutral fillers in broad concentration ranges. Only with this complexity of recipes can the high, sometimes contradictory requirements regarding friction loss, reliability, wear, and braking comfort be optimally reconciled for the majority of drivers.

Between the friction lining and the adhesive a roughly 2...3 mm thick intermediate layer similar mechanically and chemically to the friction lining can be used for example to improve the pad connection or damp braking noises.

Finally, a brake pad must be clearly marked to denote its country of use in order for example to display the regulatory approval for use in public traffic. The markings are usually printed or stamped on the backs of the brake pads.

Special regional requirements

Unlike in Europe – and in particular unlike in Germany – in many regions around the world drivers cannot drive very fast by vehicle since speed limits are enforced even on expressways and freeways. Resulting lower friction-coefficient requirements, but higher requirements with regard to wear and dust formation led for example in Asia and the USA to the development of different friction materials from those in Europe. From Japan come the so-called NAO materials (Non Asbestos Organics), which are characterized by very low brake-disk corrosion and consequently also by low dust formation. However, at present these recipes cannot deliver friction coefficients that are as high as those produced by the recipes customary in Europe. In contrast to European recipes, they typically contain no steel wool or iron powder. They are therefore also referred to as non-steel pads or more promotionally effectively as ceramics. In contrast, the materials customary in Europe are called low-steel or low-met pads. The combination of the good friction loss of European recipes with the low dust generation of NAO rec-

Table 1: Raw-material groups with typical concentration ranges of brake-pad recipes for use in Europe

Raw-material group	Raw material	Volume %
Metals	Steel wool Aluminum wool Copper wool Zinc powder	10...15
Binders	Phenolic resin Caoutchouc	15...20
Fillers	Silicates, e.g. Mica powder Talcum Chalk	20...50
Abrasives	Aluminum oxide Silicon carbide	2...5
Lubricants	Molybdenum sulfide Tin sulfide	2...10
Organic fibers	Aramid fiber Cellulose fiber	2...5
Graphite and coke	Graphite and coke	10...25

ipes constitute a current conflict of goals for the development of friction materials.

Up to half the weight of semi-met pads is made up of iron materials. They represent a good cost compromise with weaknesses in the high-load and high-temperature ranges.

Disk brakes for commercial vehicles

Special disk brakes have been developed for commercial vehicles. These disk brakes are actuated with compressed air. Because the pressure here is much smaller than that in a hydraulic brake, the brake cylinders cannot be integrated into the brake calipers. They must be flanged into position (Figure 5).

Functioning principle

Operating concept of the service brake

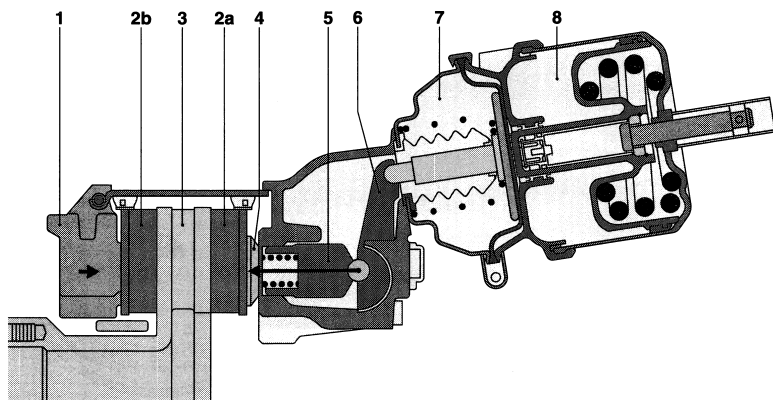
When air enters the service brake cylinder, the eccentrically mounted brake lever is actuated. The brake cylinder force is boosted by the lever ratio and transferred via the bridge and plungers to the inner brake pad. The reaction force that arises at the brake caliper is transferred by shifting the brake caliper to the outer brake pad.

Operating concept of the parking brake

When the spring-type brake cylinder is vented, the force of the pre-tensioning springs is released. Via the spring-type brake piston, this moves the piston and the push rod of the service brake cylinder to operate the brake. In the case of the parking brake, the pressure in the spring-type brake actuator is completely

Figure 5: Disk brake with combination brake cylinder

1 Brake caliper, 2a Inner brake pad, 2b Outer brake pad, 3 Brake disk, 4 Plunger, 5 Bridge, 6 Eccentrically mounted brake lever, 7 Service-brake cylinder, 8 Spring-type actuator.



eliminated, thus releasing the force of the pre-tensioning springs to achieve maximum braking effect.

Automatic self-adjusting mechanism

Disk brakes pneumatically or mechanically actuated by the spring-type actuator are fitted with automatic clearance compensation.

Wear monitoring

Continuous wear monitoring may also be provided. This is required in the case of electronic brake systems for wear adaptation and for service information systems.

Brake disks

Solid brake disks are used less frequently on commercial vehicles because they can only dissipate the heat slowly. Internally ventilated brake disks have a larger surface through which heat exchange can take place. In this design two friction rings are connected via bridges. The rotation of the brake disk creates on the inside a radial ventilation effect in the outward direction.

Brake disks for commercial vehicles are usually manufactured from gray cast iron. The carbon content increased up to the saturation limit provides good thermal conductivity.

Drum brakes

Drum brakes are radial brakes with two brake shoes. They generate their braking force on the inner friction surface of a brake drum.

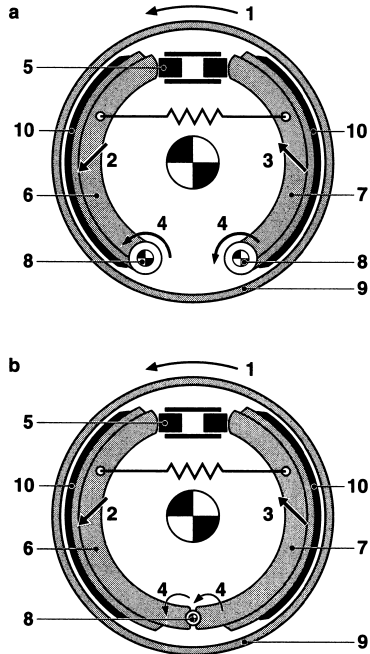
Drum-brake designs

There are two different designs of drum brake, based on how the brake shoes are guided:

- brake shoes with fixed pivot (Figures 6a and 6b),
- brake shoes as sliding shoes (Figure 7).

Figure 6: Principle of simplex brake

- a) Brake shoe with two single pivots,
b) Brake shoe with one double pivot.
1 Direction of rotation of brake drum with vehicle moving forwards,
2 Self-augmentation, 3 Self-inhibition,
4 Torque,
5 Double-acting wheel-brake cylinder,
6 Leading brake shoe (primary shoe),
7 Trailing brake shoe (secondary shoe),
8 Fulcrum (pivot),
9 Brake drum, 10 Brake lining.



At the brake shoe that rotates in the direction of the brake drum (primary shoe, leading brake shoe, Figure 6), the friction force during the braking operation creates a turning force around the brake-shoe fulcrum which in addition to the application force presses the shoe against the drum. This generates a self-augmenting effect. In a simplex brake a turning force is created around the fulcrum of the trailing brake shoe (secondary shoe) which diminishes the applied application force. This therefore creates a self-inhibiting effect.

Sliding-shoe guides are used in simplex, duplex, duo-duplex, servo and duo-servo brakes. Brake shoes with a fixed pivot are subject to unequal levels of wear in that they cannot center themselves like sliding shoes.

Principle of the simplex brake

A double-acting wheel-brake cylinder actuates the brake shoes (Figures 6a and 6b). The fulcrums of the brake shoes are pivots (two single pivots or one double pivot). When the vehicle is moving forwards, self-augmentation affects the leading brake shoe and self-inhibition affects the trailing brake shoe; the pattern is the same when the vehicle is backing up.

Principle of the duplex brake

Each brake shoe is actuated by a single-acting wheel-brake cylinder (Figure 7). The brake shoes designed as sliding shoes are supported on the back of the opposing wheel-brake cylinder. The duplex brake is single-acting, i.e., it has two leading self-augmenting brake shoes when the vehicle is moving forwards. There is no self-augmentation when the vehicle is backing up.

Simplex drum brake

Functioning principle of passenger-car brake

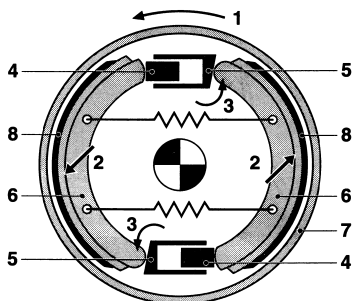
The principle of a drum brake is explained using a hydraulically operated simplex drum brake with integrated parking brake and automatic self-adjusting mechanism as an example (Figure 8). Other drum brake designs (e.g., duplex brake, duo-duplex brake) are rarely used today.

When driving extension springs pull the two brake shoes away from the brake drum so that a clearance is created between the drum friction surface and the brake linings. In the case of simplex brakes, a two-sided hydraulic wheel-brake cylinder generates the application force for the brake shoes during braking by converting the hydraulic pressure into mechanical force. Here, the leading and the trailing brake shoes with the brake pads press against the brake drums. The other ends of the brake shoes on the opposite side to the wheel-brake cylinder are braced by a support bearing that is attached to the brake anchor plate.

The leading brake shoe (primary shoe) generates a higher proportion of braking torque than the trailing brake shoe (secondary shoe). Wear is therefore greater on the primary lining. This lining is thicker or longer in design to compensate.

Figure 7: Principle of duplex brake

- 1 Direction of rotation of brake drum with vehicle moving forwards,
- 2 Self-augmentation, 3 Torque,
- 4 Wheel-brake cylinder, 5 Fulcrums,
- 6 Brake shoes, 7 Brake drum,
- 8 Brake lining.



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Functioning principle of simplex brake with S-cam for commercial vehicles

In commercial vehicles with compressed-air brake systems, the application force is frequently generated by a rotatable S-cam. S-cam rotation is effected by the brake cylinder, the brake lever (slack adjuster), and the brake-cam shaft (Figure 9).

Functioning principle of wedge-actuated brake

Wedge-actuated brakes are also used in commercial vehicles. Here, the application force for the brake shoes is generated by a wedge actuated by the brake cylinder (Figure 10).

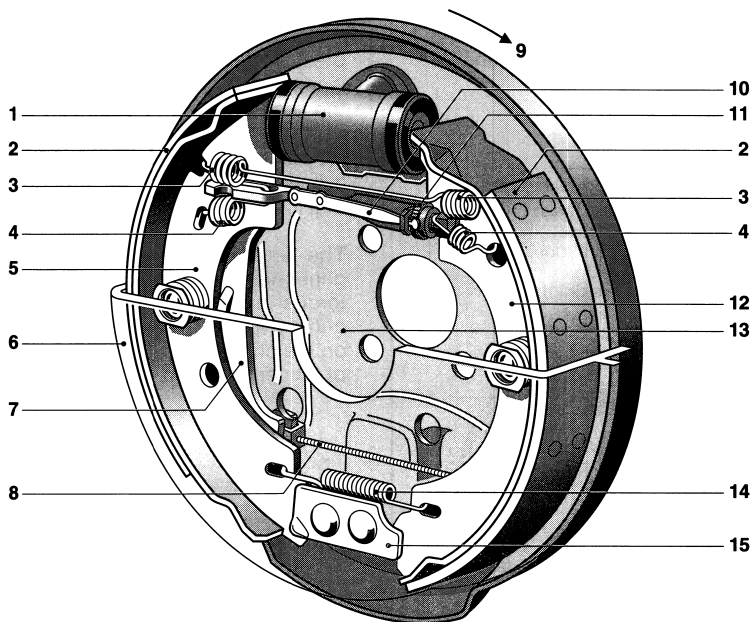
During braking the diaphragm brake cylinder is exposed to compressed air. This displaces the wedge to the right. The wedge slides between the pressure rollers. These roll on the wedge and the thrust members. The generated application force is transferred via the thrust members to the brake shoes. The excessive clearance created by brake-lining wear is compensated for by the adjusting mechanism.

Automatic adjusters

Wheel brakes must be fitted with adjusters to compensate for the increased clearance caused by lining wear. The brakes must be easily adjustable or have an au-

Figure 8: Simplex drum brake with integrated parking brake

- 1 Wheel-brake cylinder, 2 Brake lining, 3 Extension spring (for brake shoes),
- 4 Extension spring (for adjuster), 5 Trailing brake shoe,
- 6 Brake drum, 7 Parking-brake lever, 8 Brake cable,
- 9 Direction of drum rotation, 10 Thermocouple (adjuster),
- 11 Adjuster wheel (with elbow lever), 12 Leading brake shoe,
- 13 Brake anchor plate, 14 Extension spring (for brake shoes),
- 15 Brake-shoe pin bushing.



automatic adjuster (§ 41 s. 1 StVZO [1], ECE R13-H [2]).

On simplex drum brakes for passenger cars, the adjuster is part of the push rod or pressure sleeve situated under initial spring tension between the brake shoes. When the permissible clearance is exceeded, the adjuster automatically lengthens the push rod or pressure sleeve (to different extents depending on the adjuster design) and thereby adjusts the clearance between brake shoe and brake drum. Automatic adjusters mostly operate in conjunction with a thermocouple on a temperature-sensitive basis in order to prevent adjustment when the brake drum is hot (expanded).

In commercial vehicles with S-cams, the adjuster is part of the brake lever. For this purpose it has manual adjustment or is designed as an automatic linkage adjuster.

In wedge-actuated brakes, an automatically acting adjuster is integrated into the wedge mechanism.

Parking brake

A parking brake is integrated in the drum brake. The brake cable is actuated either via the handbrake lever inside the vehicle or via an electric motor with spindle. The handbrake lever is mounted at the top of the trailing brake shoe. When the parking brake is operated, the brake cable pulls the handbrake lever downwards to the right, causing the handbrake lever to press the brake shoes via the push rod against the brake drum.

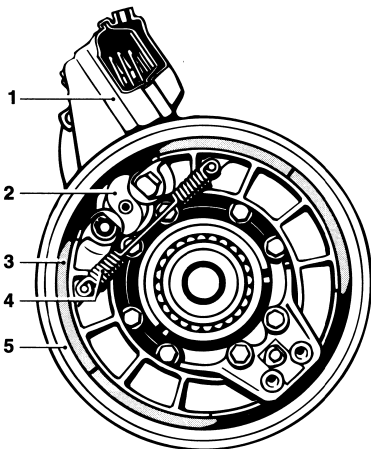
In a commercial vehicle the drum brake is actuated as a parking brake by venting the spring-type brake cylinder.

References

- [1] §41 Straßenverkehrs-Zulassungsordnung (Road traffic licensing regulations, Germany). Bremsen und Unterlegkeile (Brakes and wheel chocks). [2] ECE R13-H: Regulation No. 13-H of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of passenger cars with regard to braking. [3] B. Breuer, K. Bill (Editors): Bremsenhandbuch: Grundlagen, Komponenten, Systeme, Fahrdynamik. 5th Ed., Verlag Springer Vieweg, 2017.

Figure 9: Simplex drum brake with S-cam

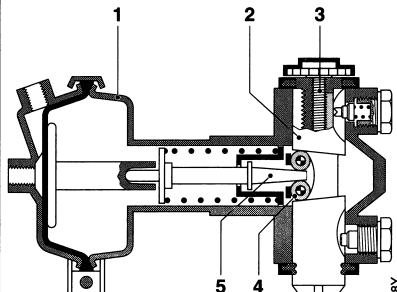
- 1 Diaphragm-type cylinder,
- 2 S-cam,
- 3 Brake shoes,
- 4 Return spring,
- 5 Brake drum.



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Figure 10: Wedge-actuated brake

- 1 Diaphragm brake cylinder,
- 2 Thrust member,
- 3 Adjusting mechanism,
- 4 Pressure rollers,
- 5 Wedge.



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Wheel-slip control systems

Function and requirements

Slip

When starting off, accelerating and braking, the efficiency required to transfer forces to the road depends on the traction available between the tires and the road surface. Slip occurs when the speed v_R at which the wheel center moves in the longitudinal direction (vehicle speed) differs from the speed v_U at which circumference rolls. Brake slip λ_B and drive slip λ_A are calculated as follows:

$$\lambda_B = \frac{v_U - v_R}{v_R}, \lambda_A = \frac{v_U - v_R}{v_U}.$$

When a wheel is locked, according to this definition brake slip $\lambda_B = -1$; when the vehicle is stationary with spinning wheels drive slip $\lambda_A = 1$ (see Slip, Fundamentals of automotive engineering).

Adhesion/slip curves

The tire must roll with slip so that it can transmit force to the road. A tire without slip would not deform on the wheel contact area and therefore could not transmit either a longitudinal force or a lateral force. The transmittable forces are dependent

on the slip. The adhesion/slip curves (Figure 2) illustrate this dependence. They progress identically for braking and propulsion/traction.

Wheel-slip control systems ensure the optimum transmission of forces between tires and road to keep the vehicle directionally stable and more easily controllable for the driver. For this purpose the longitudinal slip of the individual wheels – i.e.,

Figure 2: Adhesion/slip curve

Curve shape for dry road surface, $\mu_{HF} \approx 0.8$.

- 1 Braking or tractive force,
- 2 Lateral force.

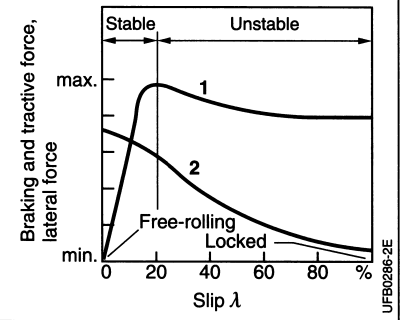
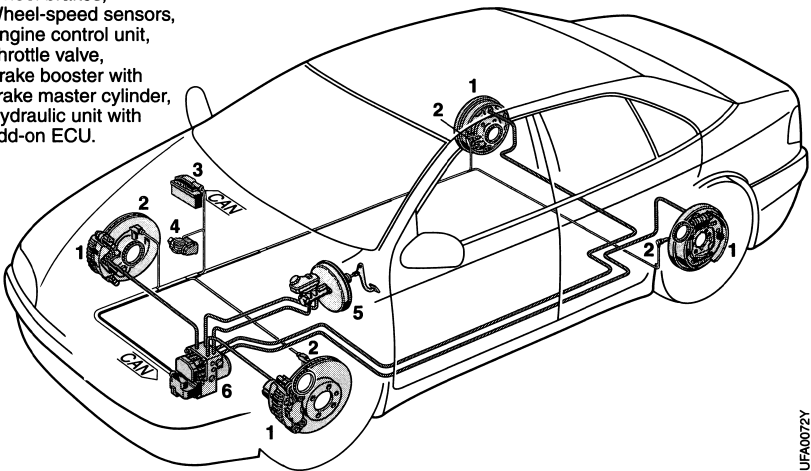


Figure 1: System diagram of a brake system with ABS

- 1 Wheel brakes,
- 2 Wheel-speed sensors,
- 3 Engine control unit,
- 4 Throttle valve,
- 5 Brake booster with brake master cylinder,
- 6 Hydraulic unit with add-on ECU.



the wheel speed referred to the velocity of the wheel center in the longitudinal direction – is regulated by modulating the braking or drive torque. There is a fundamental distinction between the antilock braking function (ABS, antilock braking system) and the traction control system (TCS).

The vast majority of acceleration and braking operations involve only limited amounts of slip, allowing response to remain within the stable range in the adhesion/slip curves. Any rise in slip (brake slip when braking and drive slip when accelerating) is accompanied initially by a corresponding increase in adhesion. Beyond this point, any further increases in slip take the curves through the maxima and into the unstable range (Figure 2) where any further increase in slip generally results in a reduction in adhesion. When braking, this results in wheel lock within a few tenths of a second. When accelerating, one or both of the driven wheels start to spin more and more as the drive torque exceeds the adhesion by an ever increasing amount.

Effect of ABS and TCS

As brake slip increases the ABS function becomes active and prevents the wheels from locking; as drive slip increases TCS prevents the wheels from spinning. Thanks to ABS the vehicle retains its directional stability and steerability even under emergency braking on a slippery road surface. The dangerous phenomenon of jackknifing is also prevented in commercial-vehicle combinations. The TCS function optimizes the transmission of forces of the drive wheels when accelerating and thereby improves both traction and stability.

Figure 1 shows an ABS system with its components for a passenger car with a hydraulic brake system. In contrast to passenger cars, commercial vehicles have pneumatic power-brake systems (air brakes). Nevertheless, the functional description of an ABS or TCS control process for passenger cars also applies in principle to commercial vehicles.

The functions of ABS and TCS have in the meantime been integrated into the driving-dynamics control.

Control systems

ABS control

Basic closed-loop control process

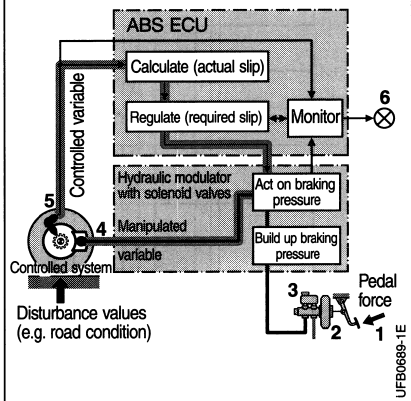
The wheel-speed sensor senses the state of motion of the wheel (Figure 3). If one of the wheels shows signs of incipient lock, there is a sharp rise in peripheral wheel deceleration and in wheel slip. If these exceed defined critical levels, the ABS controller sends commands to the solenoid-valve unit (hydraulic unit) to stop increasing or to reduce wheel brake pressure until the danger of wheel lock is averted. The braking pressure must then rise again to ensure that the wheel is not underbraked. During automatic brake control, the stability or instability of wheel motion must be detected constantly, and kept within the slip range at maximum braking force by a sequence of pressure-rise, pressure-retention and pressure-drop phases.

Typical control cycle

The control cycle depicted in Figure 4 shows automatic brake control in the case of a high friction coefficient. The change in wheel speed (braking deceleration) is calculated in the ECU. After the value

Figure 3: ABS control loop of a passenger-car system

- 1 Brake pedal, 2 Brake booster,
- 3 Brake master cylinder with fluid reservoir,
- 4 Wheel-brake cylinder,
- 5 Wheel-speed sensor,
- 6 Indicator lamp.



falls below the $(-a)$ threshold, the hydraulic-unit valve unit is switched to pressure-holding mode. If the wheel speed then also drops below the slip-switching threshold λ_1 , the valve unit is switched to pressure drop; this continues as long as the $(-a)$ signal is applied. During the following pressure-holding phase, peripheral wheel acceleration increases until the $(+a)$ threshold is exceeded; the braking pressure is then kept at a constant level.

After the relatively high $(+A)$ threshold has been exceeded, the braking pressure is increased, so that the wheel is not accelerating excessively as it enters the stable range of the adhesion/slip curve. After the $(+a)$ signal has dropped off, the braking pressure is slowly raised until, when the wheel acceleration again falls below the $(-a)$ threshold, the second control cycle is initiated, this time with an immediate pressure drop.

In the first control cycle, a short pressure-holding phase was initially necessary to filter out any faults. In the case

of high wheel moments of inertia, low friction coefficient and slow pressure rise in the wheel-brake cylinder (cautious initial braking, e.g., on black ice), the wheel might lock without any response from the deceleration switching threshold. In this case, therefore, the wheel slip is also included as a parameter in the brake-control system.

Under certain road-surface conditions, passenger cars with all-wheel drive and with differential locks engaged pose problems when the ABS system is in operation; this calls for special measures to take into account the reference speed during the control process, lower the wheel-deceleration thresholds, and reduce the engine-drag torque.

Control cycle with yaw-moment build-up delay

When the brakes are applied on a road surface with uneven grip (μ split: left-hand wheels on dry asphalt, right-hand wheels on ice), vastly different braking forces at the front wheels result and induce a turning force (yaw moment) about the vehicle's vertical axis (Figure 5).

On smaller cars, ABS must be supplemented by an additional yaw-moment build-up delay device to ensure that control is maintained during panic braking on asymmetrical road surfaces. Yaw-moment build-up delay holds back the pressure rise in the wheel-brake cylinder on the

Figure 4: ABS control cycle for high friction coefficients

v_{Ref} Reference speed,
 v_U Wheel circumferential speed,
 v_F Vehicle speed,
 a, A Wheel-deceleration thresholds.

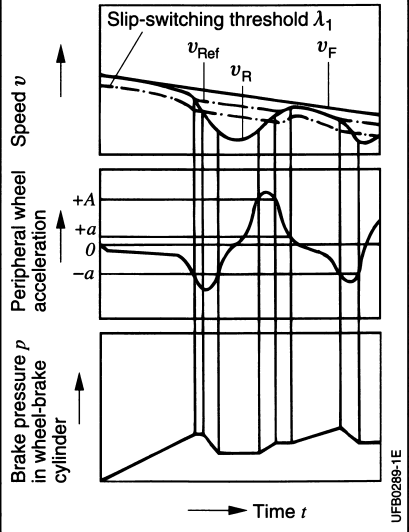
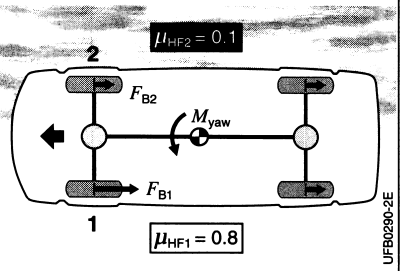


Figure 5: Yaw-moment build-up induced by large differences in friction coefficients

M_{yaw} Yaw moment, F_B Braking force,
 μ_{HF} Coefficient of friction.
1 "High wheel", 2 "Low wheel".



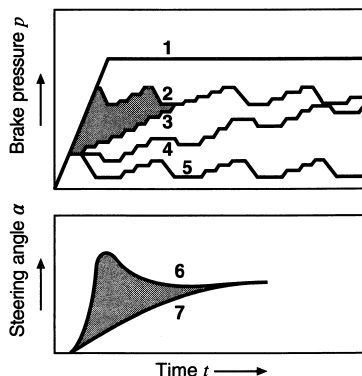
front wheel with the higher coefficient of friction at the road surface ("high wheel").

The yaw-moment build-up delay concept is demonstrated in Figure 6: Curve 1 represents the brake-master-cylinder pressure p_{MC} . Without yaw-moment build-up delay, the braking pressure at the wheel running on asphalt quickly reaches p_{high} (Curve 2), while the braking pressure at the wheel running on ice rises only to p_{low} (Curve 5); each wheel brakes with the maximum transferable braking force (see Individual control).

The yaw-moment build-up delay 1 system (Curve 3) is designed for use on vehicles with less critical handling characteristics, while yaw-moment build-up delay 2 is designed for cars which display an especially marked tendency toward yaw-induced instability (Curve 4). In all cases in which yaw-moment build-up delay comes into effect, the high wheel is under-braked at first. This means that the yaw-moment build-up delay must always be very carefully adapted to the vehicle in question in order to limit increases in stopping distances.

Figure 6: Curves for braking-pressure/steering-angle characteristic with yaw-moment build-up delay

- 1 Brake-master-cylinder pressure p_{MC} .
- 2 Brake pressure p_{high} w/o YMBD,
- 3 Brake pressure p_{high} w/ YMBD 1,
- 4 Brake pressure p_{high} w/ YMBD 2,
- 5 Brake pressure p_{low} at "low wheel",
- 6 Req. steering angle α w/o YMBD,
- 7 Req. steering angle α w/ YMBD.



Curve 6 in Figure 6 shows that for an ABS system without yaw-moment build-up delay a significantly higher steering angle is required when countersteering.

ABS control methods

The axle-based ABS control methods differ essentially in the number of control channels and the behavior when braking at μ split.

Individual control

Individual control, whereby each wheel is individually slip-controlled, produces the shortest braking distances. The drawback, however, is the yaw moment occurring under μ -split conditions, which must be compensated for by appropriate countersteering. This method is used exclusively on the rear axle since the steering and yaw moments occurring at the front axle would not be controllable for the driver when braking at μ split.

Select-low control

Select-low control (SL) is used to avoid yaw and steering moments entirely. Here, single-channel wheel-slip control is effected at the wheel with the lowest friction coefficient (select low), as a result of which both wheels on one axle receive the same brake pressure. Therefore only one single pressure-control channel per axle is required. Under μ -split conditions this produces optimum steerability and directional stability at the expense of braking distance. In the case of homogenous friction coefficients, braking distance, steerability, and directional stability are similar to those of the other methods.

Individual control, modified

"Individual control, modified" (IRM) has proven to be a good compromise between steerability, stability, and braking distance. This two-channel control method necessitates a pressure-control channel at each wheel on the axle. By appropriately limiting the brake-pressure difference between the right and left sides, the yaw and steering moments are restricted to a controllable extent. This results in a braking distance which is only a little longer than that for individual control, but it does ensure that vehicles with critical handling characteristics remain controllable.

TCS control

The traction control system has two fundamental tasks:

- optimizing traction by utilizing the available friction coefficient in the best possible way,
- ensuring vehicle stability (directional stability) by preventing the drive wheels from spinning.

To optimize traction all the driven wheels must utilize their individual friction coefficients to the maximum to the best possible extent. To this end, the wheel speeds are synchronized by active braking of the spinning wheels (brake controller or electronic differential-lock function). In this way, the braking torque exerted on the spinning wheel is available through transmission by the differential to the non-spinning wheel as drive torque.

To ensure directional stability, the wheel slip is controlled with the aid of the drive torque by the engine controller in such a way as to achieve the best possible compromise between traction and lateral stability.

Using the brake-control function described above, the driving wheels can also be synchronized so that a mechanical differential lock, if fitted, can be activated automatically, e.g., with the aid of a pneumatic cylinder. The ABS/TCS ECU calculates the correct point and conditions for releasing the differential lock.

In contrast to mechanical differential locks, the tires do not scrub on tight corners. A fundamental observation about this type of system (when it assumes an electronic brake-control function) is that it is not intended for continuous use on difficult offroad terrain. Since the brake-control function is achieved by braking the relevant wheel, brake heating is an inevitable consequence.

For multi-axle vehicles with complex drive configurations and a number of differentials (e.g., 6×4 or 8×6 with three or five differentials) the TCS function can control up to six wheels individually.

Engine drag-torque control

Particularly at low load with a low friction coefficient or very powerful engines the wheels on the powered axle can lock due

to high engine-drag torques (e.g., when downshifting), which results in unstable handling. In this case, engine drag-torque control increases the wheel speeds by increasing the drive torque and thereby prevents the impending instability. The actively exerted drive torque is limited for safety reasons.

Electronic load-dependent braking-force regulation

Load differences and the dynamic axle-load shift that occurs during sharp braking call for an adaptation of the braking forces. This was originally performed by ALB valves (automatic load-dependent braking-force regulator), which reduce the brake pressure usually on the rear axle depending on the axle load. In current ABS systems this function is assumed by electronic load-dependent braking-force regulation. Here, under minimal deceleration conditions, the differential slip between the front and rear axles is minimized, whereby the brake pressure on the rear axle is electronically reduced. This results, assuming the same friction conditions on both axles, in identical braking and consequently optimum braking under driving-dynamics considerations. This function dispenses with the need for the additional ALB valve.

ABS/TCS systems for passenger cars

The requirements imposed on an ABS system are described in the regulations ECE-R13 [1]. This regulation defines ABS as a component of a service-brake system (Figure 1) which automatically controls wheel slip in the direction of wheel rotation on one or more wheels when braked.

ECE-R13 Annex 13 defines three categories. The present generation of ABS meets the highest level of requirements (Category 1).

Components

An ABS or ABS/TCS system consists of the following components: wheel-speed sensors, electronic control unit (ECU), and hydraulic unit (for passenger cars) or pressure-control valves (for commercial vehicles).

Wheel-speed sensors

The most important input variables for controlling wheel slip are the wheel speeds, which are recorded by wheel-speed sensors. These sensors scan a rotating sensor ring and generate an electrical signal with a frequency proportional to the rotational speed (see rpm sensors).

There are basically two different types of rpm sensor: active and passive. The active rpm sensors used predominantly in passenger cars operate according to the Hall principle and can record, aside from the road-speed signal, further information such as the temperature and transmit it to the ECU.

Hydraulic unit of an ABS system

The main hydraulic components of the hydraulic unit – also called the hydraulic modulator – are the following (Figure 7):

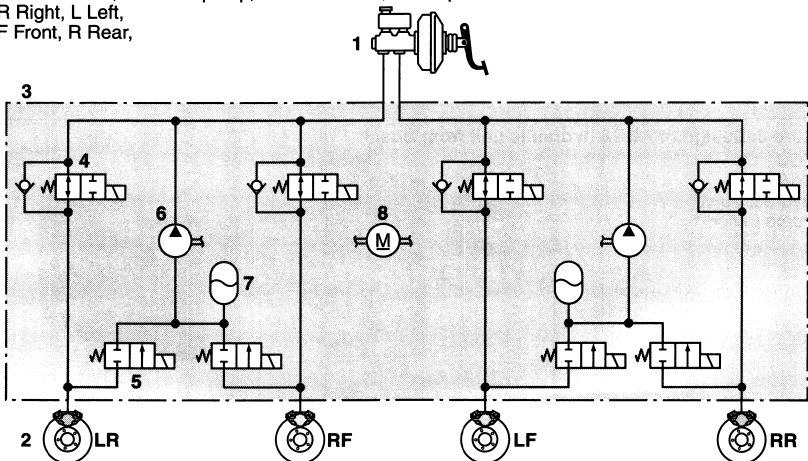
- one return pump per brake circuit,
- accumulator chamber,
- damping functions, previously performed by an accumulator chamber and a flow restrictor, are now performed both hydraulically and by control systems, i.e., software,
- 2/2-way solenoid valves with two hydraulic positions and two hydraulic connections.

There is one pair of solenoid valves for each wheel (except in the case of 3-channel configurations with front/rear brake-circuit split, see ABS system variants) – one of which is open when de-energized for pressure rise (inlet valve, IV) and one which is closed when de-en-

Figure 7: Hydraulic system of an antilock braking system.

1 Brake master cylinder, 2 Wheel-brake cylinder, 3 Hydraulic unit, 4 Inlet valve, 5 Outlet valve, 6 Return pump, 7 Accumulator, 8 Pump motor.

R Right, L Left,
F Front, R Rear,



ergized for pressure drop (outlet valve, OV). In order to achieve rapid pressure relief of the wheel brakes when the pedal is released, the inlet valves each have a non-return valve which is integrated into the valve body (e.g., non-return valve sleeves or unsprung non-return valves).

The assignment of pressure-rise and pressure-drop functions to separate solenoid valves with only one active (energized) setting has resulted in compact valve designs, i.e., smaller size and weight, as well as lower magnetic forces compared to the previous 3/3-way solenoid valves. This allows optimized electrical control with low electrical power loss in the solenoid coils and the control unit. In addition, the valve block (Figure 8) can be made smaller. This results in quite significant savings in weight and size.

The 2/2-way solenoid valves are available in a variety of designs and specifications, and, because of their compact dimensions and excellent dynamics, they allow fast electrical switching times sufficient for pulse-width-modulated cyclic operation. In other words, they have "proportional-valve characteristics".

ABS8 from Bosch (Figure 8) benefits from current-signal-modulated valve control which substantially improves function (e.g., adaptation to changes in coefficient of friction) and ease of control (e.g., smaller deceleration fluctuations with the aid of pressure stages and analog pressure control). This mechatronic optimization has positive effects not only on function but also on user-friendliness, i.e., noise and pedal feedback.

ABS 8 is capable of specific adaptation to individual vehicle-class requirements by varying the components (e.g., using motors of different power ratings, varying accumulator chamber size, etc.). The power of the return motor can vary within a range of approx. 90 to 200 watts. The accumulator-chamber size is also variable.

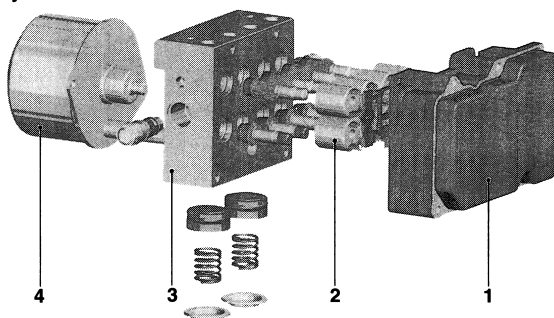
Hydraulic unit of an ABS/TCS system

On passenger cars with hydraulic brake systems, an expanded ABS hydraulic unit is required for TCS brake intervention. Depending on the variant, the expansion can comprise an intake valve and a changeover valve (Figure 9). An additional hydraulic presupply pump and a pressure accumulator may be required. During a necessary brake intervention, the intake and changeover valves assigned to the spinning wheel and the ABS return pump are electrically actuated. The return pump can draw brake fluid from the brake master cylinder through the intake valve. The changeover valve blocks the return flow to the brake master cylinder. The pressure generated by the return pump passes through the inlet valve to the wheel-brake cylinder of the spinning wheel, as a result of which the wheel is braked and prevented from spinning. Braking pressure is built up, as dictated by the situation and adapted by continuous monitoring of the control process, by alternating and electrically clocked actuation of the inlet and outlet valves in the hydraulic unit.

On completion of the control phase, electrical actuation is terminated and the braking pressure applied for TCS control,

Figure 8: Design of ABS8 hydraulic unit from Bosch

- 1 ECU,
- 2 Coil group,
- 3 Hydraulic unit,
- 4 Pump motor.



as following a normal braking operation, is reduced via the intake and changeover valves and the brake master cylinder.

Electronic control unit (ECU)

The ECU processes the signals supplied by the wheel-speed sensors. After these signals have been conditioned and filtered, a vehicle reference speed is calculated and this forms the basis for the wheel-slip calculation. To form the reference speed the individual rotational-speed signals are, depending on the respective driving situation and other criteria, corrected if necessary with different weighting.

The associated solenoid valves are activated depending on the individual wheel-slip values and the target values.

Because of the safety relevance of the slip-control functions, the ECUs contain comprehensive safety and diagnostic functions to monitor the complete system on a permanent basis. Detected faults lead to partial or full shutdown of the system and are stored in a fault memory. This can be interrogated in the garage-workshop with a diagnosis tester and deleted after the faults have been cleared.

ABS system variants

A variety of versions are available depending on the brake-circuit configuration, the vehicle's drivetrain configuration, functional requirements and cost considerations. The most common brake-circuit configuration is the diagonal split (X brake-circuit configuration), followed by the front-rear split (II brake-circuit configuration) (see Brake-circuit configuration). The HI and HH brake-circuit configurations (e.g., in the DaimlerChrysler Maybach) are specialized applications and are rarely used in combination with ABS. The ABS system variants are distinguished according to the number of control channels and wheel-speed sensors.

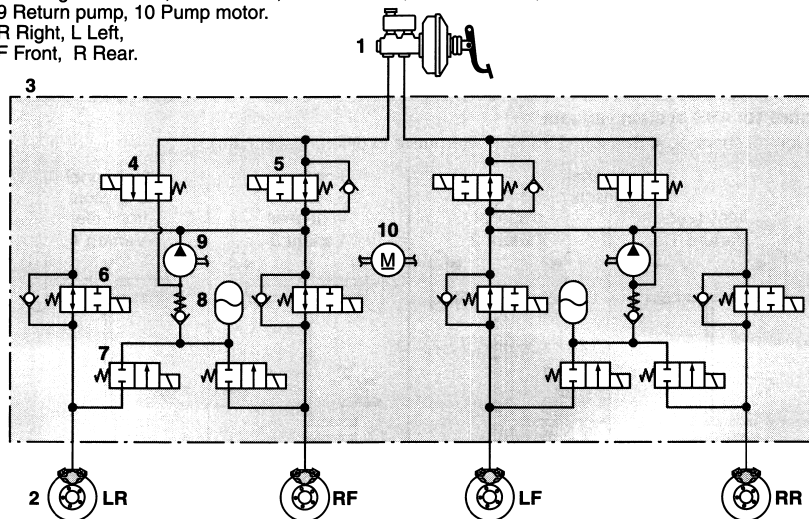
4-channel system with four sensors

These systems (Figure 10) allow individual control of the wheel brake pressure at each wheel by the four hydraulic channels, with the brake circuits split front/rear (for type-II brake-circuit configuration) or diagonally (for type-X brake-circuit configuration). Each wheel has its own wheel-speed sensor to monitor wheel speed.

Figure 9: Schematic diagram of an ABS/TCS hydraulic circuit for passenger cars with X brake-circuit configuration

1 Brake master cylinder, 2 Wheel-brake cylinder, 3 Hydraulic unit, 4 Intake valve, 5 Changeover valve, 6 Inlet valve, 7 Outlet valve, 8 Accumulator, 9 Return pump, 10 Pump motor.

R Right, L Left,
F Front, R Rear.



3-channel system with three sensors

Instead of the familiar arrangement with a separate speed sensor on each wheel, the rear wheels with this variant share a single sensor which is fitted in the differential. Due to the characteristics of the differential, it allows the measurement of wheel-speed differences with certain restrictions. Due to the select-low control characteristic for the rear wheels, i.e., parallel connection of the two rear-wheel brakes, a single hydraulic channel is sufficient for (parallel) control of the rear braking pressures.

Hydraulic 3-channel systems require a type-II brake-circuit configuration (front and rear split).

3-sensor systems can only be used on vehicles with rear-wheel drive, i.e., primarily small commercial vehicles and light trucks. The number of vehicles fitted with such systems is generally dropping off.

2-channel system with one or two sensors

Two-channel systems were produced because of the smaller number of components required and the resulting potential for cost savings. Their popularity was limited as their functionality does not match that of "full-fledged" systems. These systems are now hardly ever used in cars.

TCS engine-control intervention in passenger cars

On passenger cars with diesel engines, engine-control intervention is performed, depending on the variant, by means of electronic diesel control or the ETC (electronic throttle control) system (by reduced delivery).

On passenger cars with gasoline engines, torque reduction is usually performed by means of a combination of several functions. In this way, it is possible to reduce the engine torque according to the requirements by specifically suppressing injection pulses, retarding the ignition timing or closing the throttle device (ETC).

The engine-management systems receive the TCS request via signal or CAN data lines from the TCS control system.

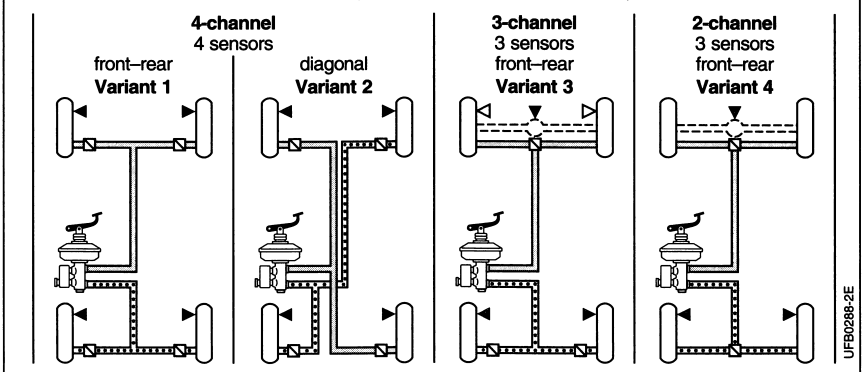
Use of ABS on motorcycles

It has been possible to reduce the size and weight of car ABS systems substantially in recent years. As a result, volume-production ABS systems are now a very attractive option for motorcycles. Consequently, this class of vehicle will also be able to benefit from the advantages of ABS as a safety system.

The car system is modified for use on motorcycles. Instead of the usual eight 2/2-way valves in the hydraulic unit for cars (with X-configuration brake circuits), motorcycles normally only require four valves. The control algorithm also differs fundamentally from that of a car ABS system.

Figure 10: ABS system variants

□ Control channel, ◀ Sensor, ◁ Sensor (alternative to differential sensor).



Other system variants have arisen from the demand for combined brake systems (CBS), i.e., systems in which both the front and rear brakes can be operated either by a foot pedal or a hand-operated lever, possibly in combination with a separate means of actuating the front brakes. This type of special case requires a 3-channel hydraulic unit. However, the CBS variant is very model-specific in design.

ABS/TCS systems for commercial vehicles

In an ABS or ABS/TCS system for commercial vehicles, depending on the vehicle configuration and the number of axles, four or six wheel-speed sensors, up to three to six pressure-control valves, and – in the case of a TCS system – one TCS valve are used (Figure 11 and Figure 12).

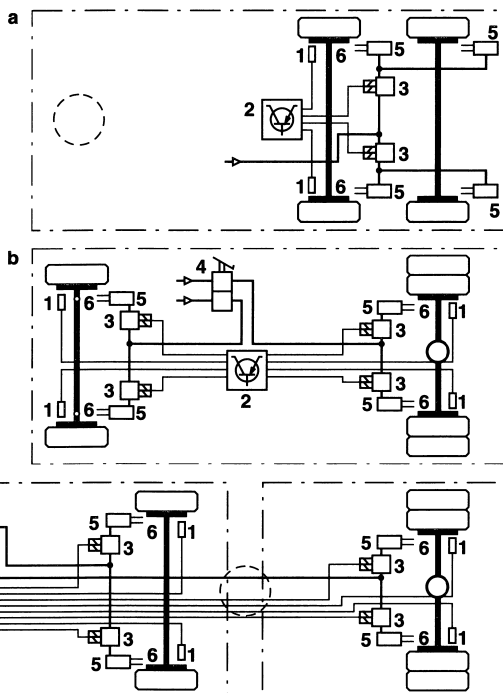
During a learning operation on initial commissioning, the control unit adjusts to the corresponding vehicle depending on the connected components. This involves detecting the number of axles, the ABS control method, and any additional functions that may be required, such as TCS.

If one axle is a lifting axle, it is automatically excluded from the ABS control process when lifted. When two axles are close together, often only one of them is fitted with wheel-speed sensors. The brake pressure of two wheels arranged

Figure 11: Examples of ABS systems for commercial vehicles

- a) Single-axle systems (semitrailers),
- b) Two-axle (four-wheel systems),
- c) Triple-axle systems (articulated bus).

- 1 Wheel-speed sensor,
- 2 ECU,
- 3 Pressure-control valve,
- 4 Service-brake valve,
- 5 Brake cylinder,
- 6 Sensor ring.



one after the other is then controlled jointly by a single pressure-control valve. In multi-axle vehicles where the axles are further apart, such as articulated buses for example, three-axle control is used out of preference.

In light commercial vehicles with pneumatic-hydraulic converters, ABS intervenes in the pneumatic brake circuit via pressure-control valves and defines the hydraulic braking pressure.

When the vehicle is running on a low-friction-coefficient road surface, the operation of an additional retarding brake (exhaust brake or retarder) can lead to excessive slip at the driven wheels. This would impair vehicle stability. ABS therefore also monitors brake slip here and controls it to permissible levels by switching the retarder on and off.

In addition, there is an ABS system independent of the tractor vehicle in the trailer, which in turn consists of two or four wheel-speed sensors and an mechatronic pressure-control module with integrated electronics.

Components

Wheel-speed sensors

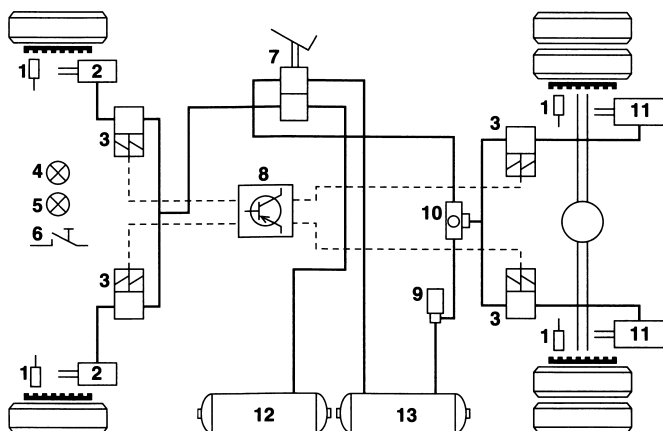
Virtually exclusively passive speed sensors in accordance with the inductive measuring principle have hitherto been used in the commercial-vehicle sector. Owing to the principles involved, these sensors can only sense speeds greater than 0 km/h, which means that – unlike active sensors, which can also detect a stationary sensor wheel – stationary detection is not possible.

Pressure-control valve

The pressure-control valves are located between the service-brake valve and the brake cylinders, and control the brake pressure of one or more wheels (Figure 13). As pilot-controlled valves, they consist of a combination of two solenoid valves and in each case a downstream pneumatic diaphragm valve, which are designed as outlet and holding valves (single-channel pressure-control valve). The electronics control the solenoid valves in the appropriate combination so that the required function is performed ("pressure holding" and "pressure drop"). If no pilot-valve actuation takes place, "pressure rise" is the result.

Figure 12: Traction control system for commercial vehicles

1 Wheel-speed sensor with sensor ring, 2 Brake cylinder, 3 ABS pressure-control valve, 4 ABS warning lamp, 5 TCS lamp, 6 TCS switch, 7 Service-brake valve, 8 ABS/TCS ECU, 9 TCS valve, 10 Shuttle valve, 11 Spring-type brake cylinder, 12 Compressed-air reservoir, circuit 1, 13 Compressed-air reservoir, circuit 2.



When braking normally (i.e., without ABS response = no incipient wheel locking), air flows through the pressure-control valves unhindered in both directions when pressure is applied to or vented from the brake cylinders. This ensures fault-free functioning of the service-brake system.

TCS valve

Usually a directly controlled TCS solenoid valve designed as a 2/2-way directional-control valve in combination with a pneumatic shuttle valve is provided for the driver-independent pressure rise for the TCS brake controller (Figure 12). In the event of a brake intervention the supply pressure is applied via the electrically actuated TCS solenoid valve and the shuttle valve to the ABS pressure-control valves, whereby the shuttle valve blocks the connection to the service-brake valve according to select-high. At the same time, on the non-active side, the holding magnet of the ABS pressure-control valve is actuated to prevent the pressure rise in the associated wheel-brake cylinder. The brake pressure can now be controlled according to the desired wheel speed with the ABS pressure-control valve on the active side.

TCS engine-control intervention in commercial vehicles

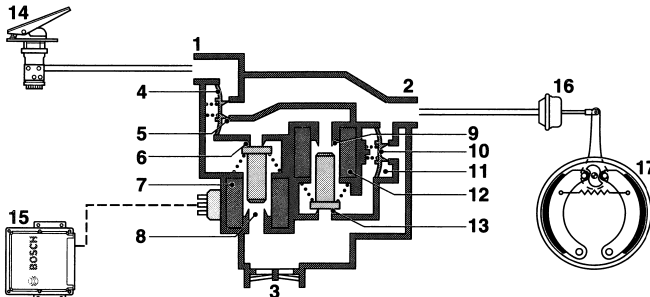
Engine-control intervention is performed – similarly to passenger cars with diesel engines – via electronic diesel control (e.g., by reduced delivery). The engine-management system receives the corresponding signal via a CAN data bus from the ABS/TCS ECU.

References

[1] ECE-R13: Uniform provisions concerning the approval of category M, N and O vehicles with regard to brakes.

Figure 13: Pressure-control valve

1 Connection for energy input, 2 Connection for energy output, 3 Venting, 4 Diaphragm, 5 Inlet, 6 Valve seat, 7 Solenoid valve for pressure-holding valve, 8 Valve seat, 9 Valve seat, 10 Diaphragm, 11 Outlet, 12 Solenoid valve for outlet valve, 13 Valve seat, 14 Service-brake valve, 15 ABS/TCS ECU, 16 Pressure-control valve, 17 Wheel brake.



Driving-dynamics control system

Function

Human error is the cause of a large portion of road accidents. Even under normal driving conditions, a driver and his vehicle can reach their physical operating limits on account of, for example, an unexpected bend in the road, a suddenly appearing obstacle or an unanticipated change in the condition of the road surface. Increased speed can also result in the driver not being able to control his vehicle safely, since the lateral-acceleration forces acting on the vehicle in such a situation reach levels which make excessive demands on him.

If the tires' coefficients of friction are exceeded, the vehicle will suddenly behave differently from what the driver with his driving experience expects. In such situations of operating limits, the driver is often no longer able to stabilize the vehicle himself; as a rule, he will intensify the instability through reactions arising from fear and panic. As a result, a significant discrepancy is built up between the longitudinal motion of the vehicle and the longitudinal axis of the vehicle (float angle β). Even by steering in the opposite direction, a normal driver will barely be able to resta-

bilize his vehicle on his own at float angles in excess of 8° .

Driving-dynamics control is known by different names – the Electronic Stability Program (ESP®, brand name of Daimler AG) or the neutral designation Electronic Stability Control (ESC). The system makes a substantial contribution to neutralizing such limit situations by helping the driver to keep his/her vehicle under control within the framework of the physical limits. Sensors constantly record the behavior of both the driver and the vehicle. By comparing the actual state with a target state appropriate for the relevant situation, the system, in the event of significant discrepancies, makes interventions in the braking system and in the drivetrain to stabilize the motion of the vehicle (Figure 1).

The integrated functionality of the anti-lock braking system (ABS) prevents the wheels from locking when the brakes are applied, while the similarly integrated traction control system (TCS) inhibits excessive wheel spin during acceleration. ESC as an overall system, however, embraces capabilities extending far beyond those of either ABS or ABS and TCS combined. The system ensures that the vehi-

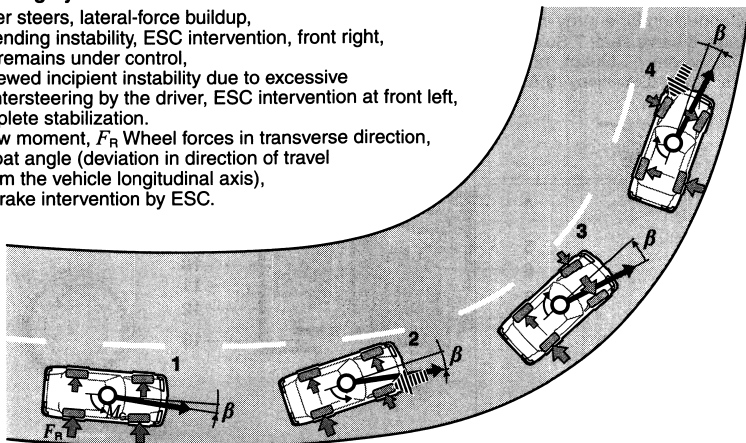
Figure 1: Transverse dynamics for a passenger car with driving-dynamics control

- 1 Driver steers, lateral-force buildup,
- 2 Impending instability, ESC intervention, front right,
- 3 Car remains under control,
- 4 Renewed incipient instability due to excessive countersteering by the driver, ESC intervention at front left, complete stabilization.

M_G Yaw moment, F_R Wheel forces in transverse direction,

β Float angle (deviation in direction of travel from the vehicle longitudinal axis),

||||| Brake intervention by ESC.



cle does not swerve outwards with its rear end (oversteering) or does not push excessively outwards with its front end (understeering), but instead follows the driver's steering input as far as is physically possible.

ESC is based on ABS and TCS components. In this way, the individual wheels can be actively braked with high dynamic response. The engine torque and thus the traction-slip values at the wheels can be influenced by means of the engine-management system. The systems communicate via, for example, the CAN bus.

Table 1: Terms and quantities

a_y	Measured vehicle lateral acceleration
F_x	Tire force in longitudinal direction
F_y	Tire force in lateral direction (lateral force)
F_N	Tire force in normal direction (normal force)
L	Distance between front and rear axles
M_{BrNom}	Nominal braking torque
$M_{DiffNom}$	Nominal differential torque
M_{EngNom}	Nominal engine torque
$M_{MWhlNom}$	Nominal sum torque
ΔM_{RedNom}	Setpoint change of engine-torque reduction
ΔM_z	Stabilizing yaw moment
p_{Whl}	Wheel-cylinder pressure
p_{Adm}	Admission pressure
r	Radius of bend
v_{ch}	Characteristic vehicle speed
v_{Dif}	Wheel-speed differential of drive wheels (on one axle)
v_{DifNom}	Nominal wheel-speed differential of drive wheels (on one axle)
v_{MWhl}	Mean wheel speed of driven axle
$v_{MWhlNom}$	Setpoint value of mean wheel speed
v_{Whl}	Measured wheel speed
v_x	Vehicle linear velocity
v_y	Vehicle lateral velocity
α	Tire slip angle
β	Float angle
δ	Steering-wheel angle
λ	Tire slip
λ_{Nom}^i	Setpoint value of tire slip at wheel i
$\Delta \lambda_{DifTolNom}$	Setpoint change of permissible slip differential of driven axle(s)
$\Delta \lambda_{Nom}$	Setpoint slip change
μ	Coefficient of friction
ψ	Yaw velocity
$\dot{\psi}_{Nom}$	Nominal yaw velocity

Requirements

Driving-dynamics control contributes to increasing driving safety. It improves vehicle behavior up to the physical operating limits. The vehicle's reaction remains foreseeable to the driver and can thus be better controlled even in critical driving situations.

At the vehicle's physical driving limits, vehicle and directional stability are enhanced in all operating states, such as full braking, partial braking, coasting, accelerating, overrunning and load changes, and also for example in the case of extreme steering maneuvers (fear and panic reactions). The risk of skidding is drastically reduced.

In a variety of different situations, further improvements are obtained in the exploitation of traction potential when ABS and TCS come into action, and when engine drag-torque control is active (automatic increase in engine speed to inhibit excessive engine braking torque). This leads to shorter braking distances and greater traction with enhanced stability and higher levels of steering response.

Incorrect system interventions could have an impact on safety. A comprehensive safety concept ensures that all faults that are not essentially avoidable are detected in time and the ESC system is shut down fully or partially depending on the type of fault.

Numerous studies (e.g. [1] and [2]) have demonstrated that ESC drastically reduces the number of accidents caused by skidding and the number of associated fatalities. The upshot of this is that ESC will become mandatory in vehicles in North America by September 2011. In the European Union (EU), all new passenger cars and light commercial vehicles have had to be fitted with a driving-dynamics control system since November 2011 (integral part of ECE-R 13-H [12]). Other new cars were subject to a transition period which must be complied with by the end of 2014. Other regions such as Japan and Australia for example will also be introducing or have already introduced such a control system.

Operating principle

Driving-dynamics control (ESC) is a system which uses a vehicle's brake system and drivetrain to deliberately influence the vehicle's longitudinal and lateral motion in critical situations. When the stability-control function assumes operation, it shifts the priorities that govern the brake system. The basic function of the wheel brakes – to decelerate and/or stop the vehicle – assumes secondary importance as ESC intervenes to keep the vehicle stable and on course. ESC can also accelerate the drive wheels by means of engine interventions to contribute to the vehicle's stability.

Both mechanisms act on the vehicle's intrinsic motion. During steady-state circular-course driving, there is a defined connection between the driver's steering input and the resulting vehicle lateral acceleration and thus the tire forces in the lateral direction (self-steering effect). The forces acting on a tire in the longitudinal and lateral directions are dependent on the tire slip. It follows from this that the vehicle's intrinsic motion can be influenced by the tire slip. The specific braking of individual wheels, e.g. of the rear wheel on the inside of the bend in the case of understeering or of the front wheel on the outside of the bend in the case of oversteering, helps the vehicle to remain on the course determined by the steering angle as precisely as possible.

Typical driving maneuver

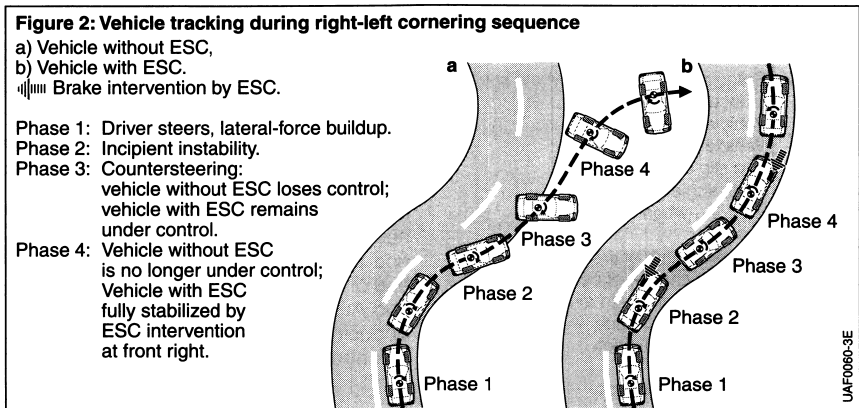
To compare how a vehicle handles at its operating limits with and without ESC, the following example is given. The driving maneuver reflects actual operating conditions, and is based on simulation programs designed using data from vehicle testing. The results have been confirmed in subsequent road tests.

Rapid steering and countersteering

Figure 2 demonstrates the handling response of a vehicle without ESC and of a vehicle with ESC negotiating a series of S-bends with rapid steering and countersteering inputs on a high-grip road surface (coefficient of friction $\mu = 1$), without the driver braking and at an initial speed of 144 km/h. Figure 3 shows the curves for dynamic-response parameters. Initially, as they approach the S-bend, the conditions for both vehicles, and their reactions, are identical. Then come the first steering inputs from the drivers (phase 1).

Vehicle without ESC

As can be seen, in the period following the initial, abrupt steering input the vehicle without ESC is already threatening to become unstable (Figure 2a, Phase 2). Whereas the steering input has quickly generated substantial lateral forces at the front wheels, there is a delay before the rear wheels start to generate similar forces. The vehicle reacts with a clockwise



movement around its vertical axis (inward yaw). The vehicle barely responds to the driver's attempt to countersteer (second steering input, Phase 3), because it is no longer under control. The yaw velocity and the side-slip angle rise radically, and the vehicle breaks into a skid (phase 4).

Vehicle with ESC

The vehicle with ESC is stabilized after the initial steering input by active braking of the front left wheel to counter the threat of instability (Figure 2b, Phase 2): This occurs without any intervention on the driver's part. This action limits the inward yaw with the result that the yaw velocity is reduced and the float angle is not subject to an uncontrolled increase. Following the change of steering direction, first the yaw moment and then the yaw velocity reverse their directions (between Phases 3 and 4). In Phase 4, a second brief brake application – this time at the right front wheel – restores complete stability. The vehicle remains on the course defined by the steering-wheel angle.

Structure of the overall system

Objective of driving-dynamics control

The control of the handling characteristics at the vehicle's physical driving limits is intended to keep the vehicle's three degrees of freedom in the plane of the road – linear velocity v_x , lateral velocity v_y and yaw velocity $\dot{\psi}$ about the vertical axis – within the controllable limits. Assuming appropriate operator inputs, driver demand is translated into dynamic vehicular response that is adapted to the characteristics of the road in an optimization process designed to ensure maximum safety.

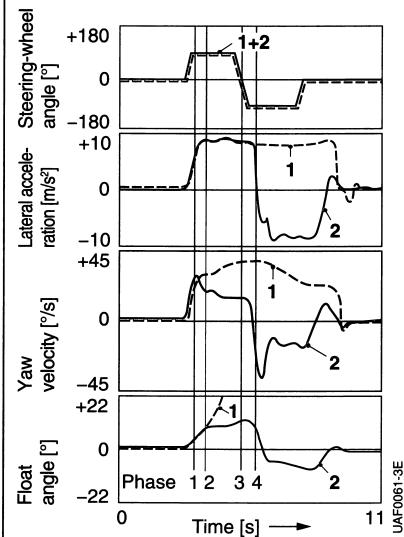
System and control structure

The ESC system comprises the vehicle as a controlled system, the sensors for determining the controller input variables, the actuators for influencing the braking, motive and lateral forces, as well as the hierarchically structured controller, comprising a higher-level transverse-dynamics controller and lower-level wheel controllers (Figure 4). The higher-level controller determines the setpoint values for the lower-level controller in the form of moments or slip, or their changes. Internal system variables that are not directly measured, such as the float angle β for example, are determined in the driving-condition estimation ("observer").

In order to determine the nominal behavior, the signals defining driver command are evaluated. These comprise the signals from the steering-wheel-angle sensor (driver's steering input), the brake-pressure sensor (desired deceleration input, obtained from the brake pressure measured in the hydraulic unit) and the accelerator-pedal position (desired drive torque). The calculation of the nominal behavior also takes into account the utilized friction-coefficient potential and the vehicle speed. These are calculated in the observer from the signals sent by the wheel-speed sensors, the lateral-acceleration sensor, the yaw sensor, and the brake-pressure sensor. Depending on the control deviation, the yaw moment, which is necessary to make the actual-state variables approach the desired-state variables, is then calculated.

Figure 3: Dynamic-response curves during a right-left cornering sequence

1 Vehicle without ESC, 2 Vehicle with ESC.
Phases 1-4 see Figure 2.



In order to generate the required yaw moment, it is necessary for the changes in desired braking torque and slip at the wheels to be determined by the transverse-dynamics controller. These are then set by means of the lower-level brake-slip and traction controllers together with the brake-hydraulics actuator and the engine-management actuator.

Driving-condition estimation

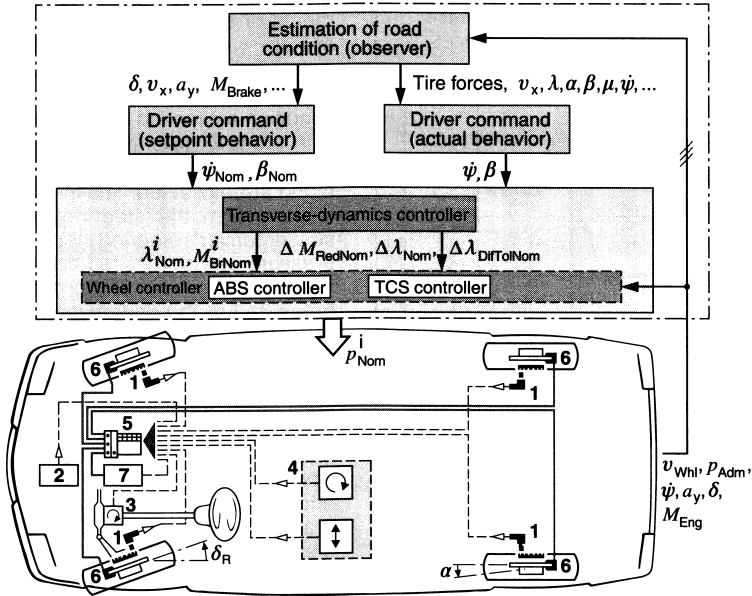
To determine the stabilization interventions, not only knowledge of the signals from the sensors for wheel speeds v_{Whl} , admission pressure p_{Adm} , yaw rate (yaw velocity) ψ , lateral acceleration a_y , steering-wheel angle δ and engine torque is important, but also knowledge of a series of further internal system variables which can be measured indirectly with appropriate effort. These include, for example, the

tire forces in the longitudinal, lateral and normal directions (F_x , F_y and F_N), the vehicle linear velocity v_x , the tire slip values λ_i , the slip angle α on one axle, the float angle β , the vehicle lateral velocity v_y , and the coefficient of friction μ . They are estimated on a model-supported basis from the sensor signals in the observer.

The vehicle linear velocity v_x is of crucial importance to all the wheel-slip-based controllers and must therefore be calculated with great accuracy. This is done on the basis of a vehicle model using the measured wheel speeds. Numerous influences must be taken into account here. The vehicle speed v_x differs for example already in normal driving situations on account of brake or drive slip from the wheel speeds v_{Whl} . For all-wheel-drive vehicles, special linking of the wheels must be taken into account in particular. During cornering, the

Figure 4: ESC overall control system

- 1 Wheel-speed sensors,
- 2 Brake-pressure sensor (integrated in hydraulic unit),
- 3 Steering-wheel-angle sensor,
- 4 Yaw sensor (yaw-rate sensor) with integrated lateral-acceleration sensor,
- 5 ESC hydraulic unit (hydraulic modulator) with mounted ECU,
- 6 Wheel brakes, 7 Engine ECU.



wheels on the inside of the bend follow a different course from the wheels on the outside of the bend, and consequently rotate at a different speed.

Vehicle handling changes during normal use in response to varying load, altered tractive resistance (e.g. road gradient or surface, wind) or wear (e.g. of the brake pads).

Under all these boundary conditions, the vehicle linear velocity must be estimated with a deviation of a few % in order to ensure the enabling and intensity of stabilization interventions to the necessary extent.

Basic transverse-dynamics controller

The function of the transverse-dynamics controller is to calculate the actual behavior of the vehicle from, for example, the yaw-velocity signal and the float angle estimated in the observer, and to bring the driving behavior in the driving-dynamic limit range into line with behavior in the normal range as closely as possible (nominal or setpoint behavior).

The connection that exists during steady-state circular-course driving between the yaw velocity and the steering-wheel angle δ , the vehicle linear velocity v_x and characteristic vehicle variables are used to determine the nominal behavior. The single-track vehicle model (see e.g. [3]) is used to produce

$$\dot{\psi} = \frac{v_x}{l} \delta \frac{1}{1 + \left(\frac{v_x}{v_{ch}}\right)^2}$$

as the basis for calculating the vehicle nominal motion. In this formula, l denotes the distance between the front and rear axles. Geometric and physical parameters of the vehicle model are summarized in the "characteristic vehicle speed" v_{ch} .

The variable $\dot{\psi}$ is then limited according to the current friction-coefficient conditions and to the special properties of the vehicle dynamics and the driving situation (e.g. braking or acceleration by the driver) and to the particular conditions such as a sloping road surface of different friction coefficients under the vehicle (μ split). The driver command is thus known as the nominal yaw velocity $\dot{\psi}_{Nom}$.

The transverse-dynamics controller compares the measured yaw velocity with the associated setpoint value and in the event of significant deviations calculates the yaw moment that is required to match the actual state variable to its setpoint state. At a higher level, the float angle β is monitored and, as the values rise, increasingly taken into consideration in the calculation of the stabilizing yaw moment ΔM_z . This controller output variable is applied by means of braking-torque and slip inputs to the individual wheels which must be adjusted by the lower-level wheel controllers.

Stabilization interventions are performed at the wheels, the braking of which generates a yaw moment in the required direction of rotation and at which the limit of the transmittable forces has not yet been reached. For an oversteering vehicle, the physical limit is first exceeded on the rear axle. Stabilization interventions are therefore performed via the front axle. For an understeering vehicle, the situation is reversed (see e.g. [6]).

The nominal slip values requested by the transverse-dynamics controller λ_{Nom}^i at individual wheels are set with the aid of lower-level wheel controllers (see Figure 4). A distinction is made between the following three application cases.

Wheel control in the coasting case

In order to exert as accurately as possible the yaw moments required to stabilize the vehicle, the wheel forces must be altered under defined conditions by controlling the wheel slip. The nominal slip requested by the transverse-dynamics controller at a wheel is adjusted in the unbraked case by the lower-level brake-slip controller by way of an active pressure build-up. The current slip at the wheel must be known as precisely as possible for this purpose. This is calculated from the measured wheel-speed signal and the vehicle linear velocity determined in the observer v_x . The nominal braking torque at the wheel is formed from the deviation of the actual wheel slip from its setpoint value using a *PID* control law.

It is not only in the event of an active pressure build-up by transverse-dynamics control that a wheel can be subject to

brake slip. Following downshifts and when the accelerator is suddenly released, inertia in the engine's moving parts exerts a degree of braking force at the drive wheels. Once this force and the corresponding reactive torque rise beyond a certain level, the tires will lose their ability to transfer the resulting loads to the road and will tend to lock (e.g. because the road is suddenly slippery). The brake slip for the driven wheels can be limited in the coasting case by engine drag-torque control. This acts like "gentle acceleration" by the driver.

Wheel control in the braked case

In the braked case, different actions overlap at individual wheels, depending on the driving situation:

- driver input via the brake pedal and the steering wheel,
- effect of the ABS controller, which prevents individual wheels from locking,
- interventions of the transverse-dynamics controller which ensures vehicle stability by specifically braking individual wheels if necessary.

These three requirements must be coordinated in such a way that the driver's deceleration and steering inputs are implemented as much as possible. If wheel control is performed primarily with the objective of maximum vehicle deceleration, it can be performed on the basis of wheel acceleration which can be robustly determined with minimal sensor information (instability control). In order to specifically adjust the longitudinal and lateral tire forces to stabilize the vehicle, the principle of slip control [4] must be applied because it also permits wheel control in the unstable range of the friction-coefficient/slip characteristic. From the available sensor information, however, it must be possible to determine the absolute wheel slip to a few %, depending on the vehicle speed.

The function of the ABS controller is to ensure vehicle stability and steerability in all road conditions and in so doing exploit as much as possible the friction between wheels and road. It does this also in its capacity as the lower-level controller to

the transverse-dynamics controller by modulating the brake pressure at the wheel in such a way that the maximum possible longitudinal force can be exerted while maintaining sufficient lateral stability. However, more variables are measured in ESC than in a pure ABS configuration, which only contains the wheel-speed sensors. Thus, individual vehicle-motion information, such as for example yaw rate or lateral acceleration, is available through direct measurement of greater accuracy than is the case with model-supported estimation of the basis of few measured values.

In certain situations, it is possible to increase performance by adapting ABS control by means of inputs from the transverse-dynamics controller. When a vehicle decelerates on unequal road surfaces (μ split), very different braking forces occur at the wheels on the left and right sides of the vehicle. This generates a yaw moment about the vehicle vertical axis, to which the driver must react by countersteering in order to stabilize the vehicle. How quickly this yaw moment is built up – and how fast the driver must consequently react – depends on the vehicle's moment of inertia about the vertical axis. ABS features yaw-moment build-up delay to hold back the pressure rise at the front wheel with the higher coefficient of friction at the road surface ("High" wheel). This ABS facility can also use information from the higher-level transverse-dynamics controller (on the driver's reaction and the vehicle behavior) and thereby react even better to the actual vehicle motion.

If, when braking in a bend, the vehicle starts to turn under certain conditions, the oversteer tendency can be counteracted by means of electronic braking-force distribution through pressure reduction in individual wheels. If this is not sufficient on its own, the transverse-dynamics controller helps by actively building up pressure at the front wheel on the outside of the bend (reduction of lateral force). If, on the other hand, the vehicle understeers, the braking torque is increased at the rear on the inside (provided the wheel is not subject to ABS control) and decreased slightly at the front on the outside.

If the vehicle starts to oversteer during a fully or partially braked lane change, the pressure at the rear wheel on the inside of the bend is specifically decreased (increase in lateral force) and the pressure at the front wheel on the outside of the bend increased (decrease in lateral force). If the vehicle understeers while braking in the bend, the braking torque is increased at the rear on the inside (provided the wheel is not yet in the ABS control range) and decreased slightly at the front on the outside.

Wheel control in the drive case

The lower-level traction controller (TCS) is activated as soon as the drive wheels start to spin in the drive case. The measured wheel speed and the respective drive slip can be influenced by changing the torque balance at each drive wheel. The TCS controller limits the drive torque at each drive wheel to the drive torque that can be transferred there to the road. In this way, the driver command is implemented after acceleration as well as is physically possible and, at the same time, fundamental directional stability is ensured, since the lateral forces at the wheel are not too greatly reduced.

In a vehicle with a powered axle, the mean wheel speed of the driven axle

$$v_{MWhl} = \frac{1}{2} (v_{Whl}^L + v_{Whl}^R)$$

and the wheel-speed differential

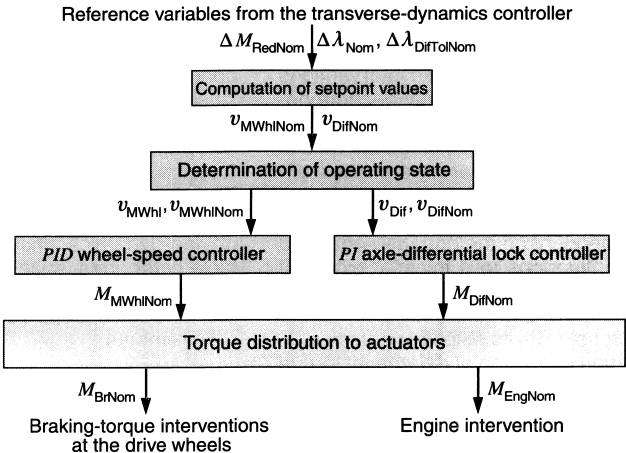
$$v_{Dif} = v_{Whl}^L - v_{Whl}^R$$

between the measured wheel speed of the left wheel v_{Whl}^L and the right wheel v_{Whl}^R are used as controlled variables.

The structure of the entire TCS controller is depicted in Figure 5. The reference variables of the transverse-dynamics controller are included in the setpoint-value calculation for the mean wheel speed and the wheel-speed differential, as well as the setpoint slip values and the coasting wheel speeds. In the calculation of the setpoint values v_{DifNom} (setpoint wheel-speed differential of the drive wheels on one axle) and $v_{MWhlNom}$ (setpoint value of the mean wheel speed), the inputs for changing the setpoint slip $\Delta\lambda_{Nom}$ and the permissible slip differential $\Delta\lambda_{DifTolNom}$ of the drive axle(s) act in the form of an offset on the basic values calculated in the TCS. In addition, an understeer or oversteer tendency identified by the transverse-dynamics controller directly influences, through the setpoint change of the en-

Figure 5: Structure of TCS controller

Quantities, see Table 1.



gine-torque reduction ΔM_{RedNom} , the determination of the maximum permissible drive torque.

The dynamic response of the drivetrain depends on the highly differing operating states. It is therefore necessary to determine the current operating status (e.g. selected gear, clutch actuation) in order to be able to adapt the controller parameters to the controlled system's dynamic response and to nonlinearities.

Because the mean wheel speed is affected by variable inertial forces originating from the drivetrain as a whole (engine, transmission, drive wheels, and the propshaft itself), a relatively large time constant is employed to describe its correspondingly leisurely rate of dynamic response. The mean wheel speed is controlled by means of a nonlinear *PID* controller, whereby in particular the gain of the *I*-component (dependent on the operating status) can vary over a wide range. In the stationary case, the *I*-component is a measure for the torque which can be transferred to the road surface. The output variables of this controller is the setpoint sum torque M_{MWhlNom} .

In contrast, the time constant for the wheel-speed differential is relatively small, reflecting the fact that the wheels' own inertial forces are virtually the sole determining factor for their dynamic response. Furthermore, in contrast to the mean wheel speed, it is influenced only indirectly by the engine. The wheel-speed differential v_{Diff} is controlled by a nonlinear *PI* controller. Because brake interventions at a drive wheel initially only become noticeable through the torque balance of this wheel, they change the distribution ratio of axle differential and thereby emulate a differential lock. The controller parameters of this axle-differential-lock controller are only dependent on the engaged gear and engine influences to a minimal extent. If the differential speed on the driven axle deviates more than currently permissible (dead zone) from its setpoint value v_{DiffNom} , calculation of a setpoint differential torque M_{DiffNom} starts. The dead zone is widened if TCS brake interventions are to be avoided, for example when cornering at the operating limits.

Setpoint sum and setpoint differential torques are the basis for distributing the positioning forces to the actuators. The setpoint differential torque M_{DiffNom} is set by the braking-torque difference between the left and right drive wheels by means of corresponding valve actuation in the hydraulic unit (asymmetrical brake intervention). The setpoint sum torque M_{MWhlNom} is adjusted by both the engine interventions and a symmetrical brake intervention.

With a gasoline engine, adjustments undertaken through the throttle valve are relatively slow to take effect (lag and the engine's transition response). Retarding the ignition timing and, as a further option, selective suppression of injection pulses are employed for rapid engine-based intervention. In diesel-engine vehicles, the electronic diesel control system (EDC) reduces the engine torque by modifying the quantity of fuel injected. Symmetrical brake intervention can be applied for brief transitional support of engine-torque reduction.

Traction plays a special role in off-road applications. Normally, in vehicles with off-road requirements, traction control is automatically adapted by way of special situation identification in order to achieve the best levels of performance and robustness possible. Other vehicle manufacturers give the driver the opportunity to choose different adjustments from deactivation of engine-torque limitation through to adaptations tailored to special road conditions (e.g. ice, snow, grass, sand, slush, and rocky ground).

Supplementary transverse-dynamics functions

The basic ESC functions described above can also feature supplementary driving-dynamics functions for special vehicle categories, such as sport utility vehicles (SUVs) and small vans for example, and for special vehicle-stabilization requirements.

Enhanced understeering control

It is possible even in normal driving conditions for the vehicle to fail to comply adequately with the driver's steering input

(it understeers) if, for example, the road surface in a bend is suddenly wet or contaminated. ESC can therefore increase the yaw rate by exerting an additional yaw moment. This enables the vehicle to negotiate a bend at the maximum speed physically possible. The expected frequency of interventions and the comfort requirements of the vehicle manufacturer differ from vehicle type to vehicle type, and there are accordingly different expansion stages for executing such brake interventions which influence the vehicle's understeering behavior.

If the driver requests a smaller radius of bend than is physically possible, then only the reduction of the vehicle speed remains. This can be read from the connection applicable during steady-state cornering between the radius of bend r , the vehicle linear velocity v_x and the yaw rate $\dot{\psi}$:

$$r = \frac{v_x}{\dot{\psi}}.$$

In order to ensure a desired track course, the vehicle is then – without a yaw moment being applied – braked as far as necessary by specific braking of all the wheels (Enhanced Understeering Control, EUC).

Rollover prevention

In particular, light commercial vehicles and other vehicles with a high center of gravity, such as sport utility vehicles (SUVs), can overturn when high lateral forces are generated by a spontaneous steering reaction by the driver in the course of an evasive maneuver on a dry road for example (highly dynamic driving situations) or when the lateral acceleration of a vehicle slowly increases into the critical range as it negotiates a freeway exit with a decreasing radius of bend at excessive speed (quasi-stationary driving situations).

There are special functions (Rollover Mitigation Functions, RMF) which identify these critical driving situations by using the normal ESC sensors and stabilize the vehicle by intervening in brake and engine control. In order to ensure intervention on time, in addition to the driver's steering

input and the measured reaction of the vehicle (yaw rate and lateral acceleration), a predictive process is used to estimate the vehicle's behavior in the near future. The two wheels on the outside of the bend, in particular, are braked if an imminent danger of overturning is identified. This action reduces the lateral forces on the wheels and thereby reduces the critical lateral acceleration. Particularly in the event of highly dynamic evasive maneuvers, wheel control must be effected with such high levels of sensitivity that, in spite of the wildly fluctuating vertical forces F_N , vehicle steerability is not diminished by the tendency of individual wheels to lock. The reduction of wheel speed by individual wheel braking also ensures that the driver is able to keep the vehicle in lane. In quasi-stationary driving situations, punctual reduction of the engine torque also prevents the driver from provoking a critical situation.

The moment of intervention and the intensity of the stabilizing interventions must be adapted as accurately as possible to the current vehicle behavior. This behavior can change significantly with the load, for example in the case of vans or sport utility vehicles fitted with roof racks. Such vehicles therefore make use of additional estimation algorithms which calculate the vehicle mass and the change in the center of mass caused by load distribution, if this is required to adapt the ESC functions (Load Adaptive Control, LAC).

Trailer sway mitigation

Depending on the vehicle speed, combinations of towing vehicle and trailer are prone to swaying about their vertical axis. If the vehicle is traveling at a slower speed than the "critical speed" (normally between 90 km/h and 130 km/h), these swaying motions are adequately damped and are quick to die down. If, however, the combination is traveling at a higher speed, small steering movements, crosswinds or driving over a pothole can suddenly induce such swaying motions, which then quickly intensify and can ultimately cause an accident due to the combination jackknifing.

Clear periodic oversteering triggers normal ESC stabilization interventions, but these normally arrive too late and on their own are not sufficient to stabilize the combination. The Trailer Sway Mitigation (TSM) function identifies swaying motions in good time on the basis of the customary ESC sensors; it does this by model-based analysis of the towing vehicle's yaw rate while taking into account the driver's steering movements. When these swaying motions reach a critical level, the combination is automatically braked in order to reduce the speed to such an extent that not even the smallest subsequent excitation will cause an immediate critical oscillation again. In order to damp the oscillation as effectively as possible in a critical situation, in addition to symmetrical deceleration through all the towing vehicle's wheels, individual wheel interventions are performed which swiftly damp the swaying motion of the combination. Limitation of the engine torque prevents dangerous acceleration by the driver during the stabilization process.

Activation of further driving-dynamics actuators

In addition to utilization of the hydraulic wheel brakes, other actuators are provided by means of which the driving-dynamic properties of a vehicle can be specifically influenced. When active steering and chassis systems are linked with ESC to form the composite system known as Vehicle Dynamics Management (VDM), they can, in their entirety, support the driver even better and thereby improve safety and driving dynamics even further.

While the combination of the steering or roll-stabilization system with the brake system has been introduced in the last few years [5], systems for activating differential locks in the drivetrain have been established on the market for some time now. The large number of such systems means that linking with ESC is possible in many cases. The supplementary actuator can basically be activated either directly from the extended ESC function (cooperation approach) or via a separate ECU which exchanges information with the ESC ECU (coexistence approach).

Figure 6: Drive concept of an all-wheel-drive vehicle with ESC

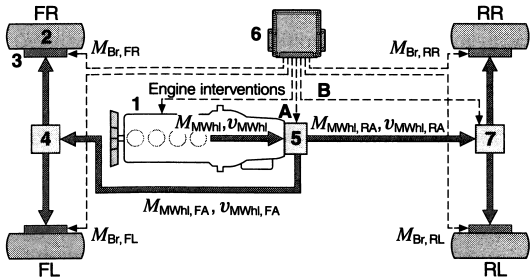
- 1 Engine with transmission,
- 2 Wheel,
- 3 Wheel brake,
- 4 Axle differential,
- 5 Central differential,
- 6 ECU with enhanced ESC function,
- 7 Axle differential.

Engine, transmission, gear ratios of differentials and losses are combined into one unit.

- A Lock interventions with active central differential,
- B Torque-vectoring interventions.

- v Wheel speed,
- v_{MWhl} Mean wheel speed,
- M_{MWhl} Driving sum torque,
- M_{Br} Braking torque,
- FA Front axle
- RA Rear axle

- R Right, L Left,
- F Front, R Rear,
- FA Front axle,
- RA Rear axle.



In all-wheel-drive vehicles, the drive torque is distributed via a central element to both powered axles (Figure 6). When the engine acts first and foremost on one axle and the second axle is linked via the central element, this is known as a hang-on system. If this central element is an open differential (without a locking action), drive torque is limited when one axle demonstrates increased slip. In the most unfavorable case, propulsion cannot be achieved if a wheel spins. In combination with ESC, symmetrical brake interventions by the all-wheel TCS controller can limit the differential speed between the axles and thereby achieve a longitudinal locking action.

The traction control of ESC can also be matched to the special operating concept of other types of central elements such as Torsen and viscous couplings. Basically, all the controllable drivetrain actuators must demonstrate a defined locking moment and dynamic response when opening and closing in order to specifically adapt the vehicle's self-steering properties with them.

If the drivetrain of a vehicle can be manually switched over between different configurations, ESC can be automatically adjusted to the operating mode selected by the driver. Because ESC is based on individual wheel control, cooperation with mechanical differential locks for specific off-road conditions is only possible if the differential lock can be automatically opened during interventions by the transverse-dynamics controller. The system must otherwise be switched to an ABS fallback level when the lock is engaged, because driving-dynamics interventions at one wheel would also affect other wheels if the axles were rigidly linked.

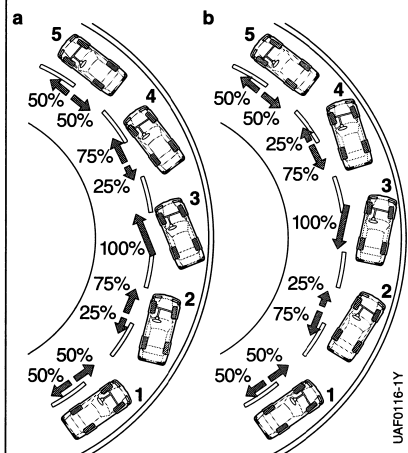
In addition to simple links between the two axles, there are controllable central locks in which an electric or hydraulic actuator activates a coupling and thereby adapts the locking moment (Figure 6, A). In this way, it is possible with the ESC driving-dynamics information (e.g. wheel speeds, vehicle speed, yaw rate, lateral acceleration, and engine torque) and by also taking into account actuator-specific variables (such as e.g. the mechanical load on the component) to optimally adapt the linking of the two axles to the

current driving situation (Dynamic Coupling Torque at Center, DCT-C).

The example in Figure 7 shows how variable drive-torque distribution influences the vehicle behavior. If, in the event of a risk of oversteering in a bend, provisionally more drive torque can be shifted to the front axle, it is necessary only much later to avoid instability, to lower the engine torque or even stabilize the vehicle with brake interventions (the maximum possible drive-torque shift is shown). If a vehicle tends to understeer, this tendency can be lessened by shifting drive torque to the rear axle. In both cases, vehicle behavior with improved response and better stability is achieved. The limits within which shifting the drive torque is actually possible are dependent on the concrete drivetrain configuration.

Figure 7: Influence of drive-torque distribution on vehicle behavior

- a) Oversteer: Stability limit is first exceeded on the rear axle,
 - b) Understeer: Stability limit is first exceeded on the front axle.
- 1 Standard distribution during stable driving,
 - 2 Incipient instability, drive torque is shifted to the axle which still has stability potential,
 - 3 Maximum shift of drive torque,
 - 4 Withdrawal of shift,
 - 5 Standard distribution is re-established after instability has been reduced.



A controllable element on one axle can be activated by ESC along similar lines to the flexible linking of the two axles described. In terms of basic operation, the Dynamic Wheel Torque Distribution (DWT) function barely differs from the axle-differential lock effected by TCS via the hydraulic wheel brakes. However, such a supplementary actuator also actively distributes the drive torque between the wheels on one axle in normal situations. This is done with minimal losses and which much greater sensitivity and comfort than can be achieved by traction control in combination with braking-torque control and engine-torque reduction while taking into account the wear of the ESC hydraulic unit.

System components

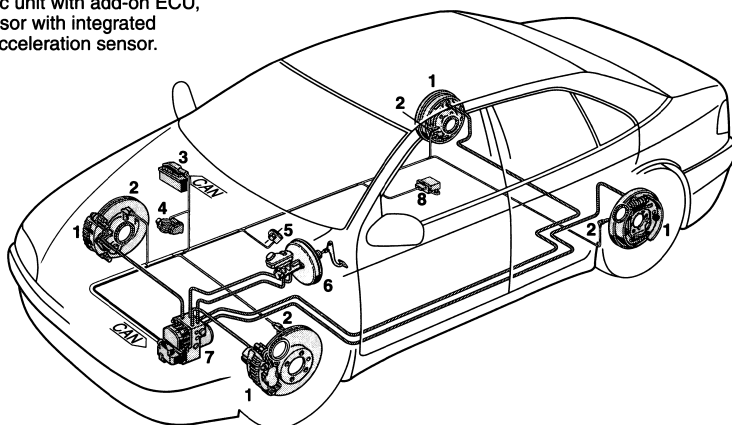
The hydraulic unit, the ECU directly connected to it (add-on ECU) and the speed sensors are suitable for the rugged ambient conditions that are encountered in the engine compartment or the wheel arches. The yaw sensor and the lateral-acceleration sensor are either integrated into the ECU or like the steering-angle sensor installed in the passenger cell. Figure 8 shows by way of example the installation locations of the components in the vehicle with the electrical and mechanical connections.

Electronic control unit (ECU)

The ECU of printed-circuit board design comprises, as well as a dual-core computer, all the drivers and semiconductor relays for valve and pump activation, as well as interface circuits for sensor-signal conditioning and corresponding switch inputs for supplementary signals (e.g. brake-light switch). There are also interfaces (CAN, FlexRay) for communicating with other systems, such as engine and transmission management for example.

Figure 8: ESC components

- 1 Wheel brakes,
- 2 Wheel-speed sensors,
- 3 Engine ECU,
- 4 Electronically activated throttle valve,
- 5 Steering-angle sensor,
- 6 Brake booster with brake master cylinder,
- 7 Hydraulic unit with add-on ECU,
- 8 Yaw sensor with integrated lateral-acceleration sensor.



Hydraulic unit

The hydraulic unit (also called the hydraulic modulator), as in ABS or ABS/TCS systems, forms the hydraulic connection between the brake master cylinder and the wheel-brake cylinders. It converts the control commands of the ECU and uses solenoid valves to control the pressures in the wheel brakes. The hydraulic circuit is completed by bores in an aluminum block. This block is also used to accommodate the necessary hydraulic function elements (solenoid valves, plunger pumps and reservoir chambers).

ESC systems require twelve valves irrespective of the brake-circuit configuration (Figure 9). In addition, a pressure sensor is usually integrated which measures the driver's deceleration command by way of the brake pressure in the brake master cylinder. This increases the performance of vehicle stabilization in partially active maneuvers. Pressure is modulated during ABS control (passive control) using ESC hydraulics in the same way as described for the ABS system.

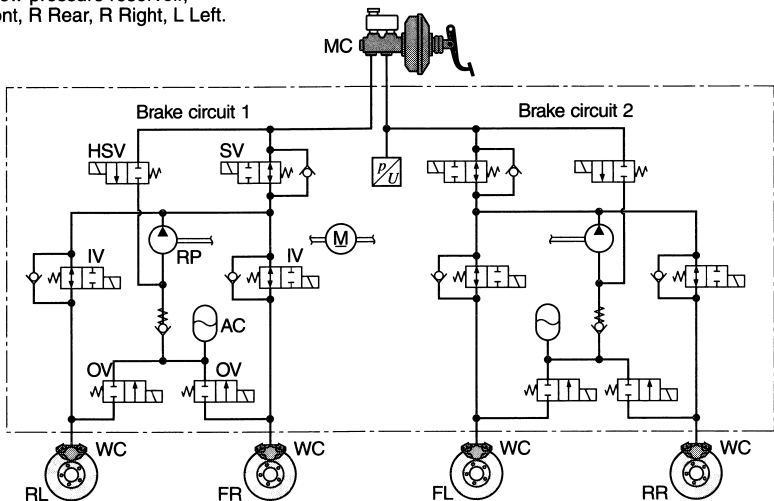
But because ESC systems must also actively build up pressure (active control) or increase a brake pressure input by the driver (partially active control), the return pump used for ABS is replaced by a self-priming pump for each circuit. The wheel-brake cylinders and the brake master cylinder are connected via a changeover valve open at zero current and a high-pressure switching valve.

An additional non-return valve with a specific closing pressure prevents the pump from drawing unwanted brake fluid from the wheels. These pumps are driven by a DC motor based on demand. The motor drives an eccentric bearing located on the shaft of the motor.

Three examples of pressure modulation are shown in Figure 10. In order to build up pressure independently of the driver (Figure 10c), the switchover valves are closed and the high-pressure switching valves opened. The self-priming pump now pumps brake fluid to the relevant wheel or wheels in order to build up pressure. The inlet valves of the other wheels

Figure 9: Hydraulic circuit diagram of an ESC hydraulic unit (X-brake-circuit configuration)

MC Brake master cylinder, WC Wheel-brake cylinder,
IV Inlet valve, OV Outlet valve, SV Switchover valve,
HSV High-pressure switching valve,
RP Return pump, M Pump motor,
AC Low-pressure reservoir,
F Front, R Rear, R Right, L Left.

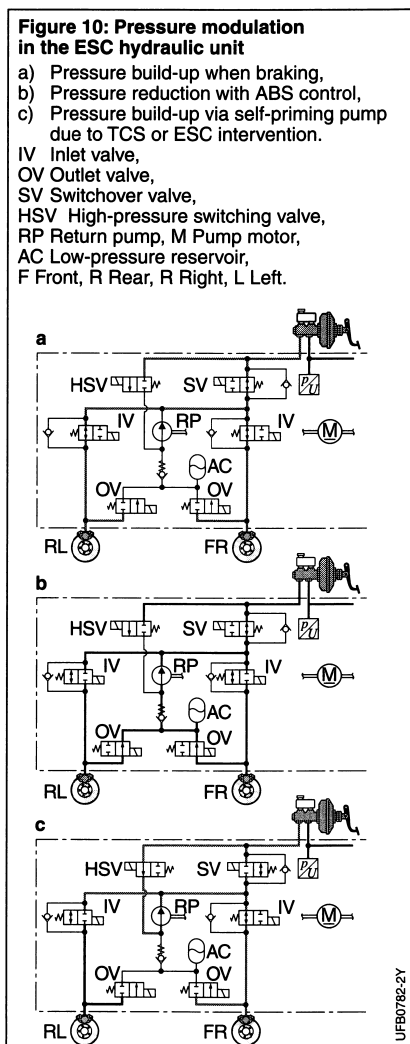


remain closed. To reduce the pressure, the outlet valves are opened and the high-pressure switching valves and switchover valves return to their original position (Figure 10b). The brake fluid escapes from the wheels into the low-pressure reservoirs, which are run empty by the pumps. Demand-based control of the pump motor reduces noise emission during pressure generation and regulation.

Figure 10: Pressure modulation in the ESC hydraulic unit

- Pressure build-up when braking,
- Pressure reduction with ABS control,
- Pressure build-up via self-priming pump due to TCS or ESC intervention.

IV Inlet valve,
OV Outlet valve,
SV Switchover valve,
HSV High-pressure switching valve,
RP Return pump, M Pump motor,
AC Low-pressure reservoir,
F Front, R Rear, R Right, L Left.



For partially active control (Figure 10a), the high-pressure switching valve must be able to open the suction path of the pump against higher differential pressures (> 0.1 MPa). The first stage of the valve is opened via the magnetic force of the energized coil; the second stage via the hydraulic area difference. If the ESC controller detects an unstable vehicle state, the switchover valves (open at zero current) are closed and the high-pressure switching valve (closed at zero current) is opened. The two pumps then generate additional pressure in order to stabilize the vehicle. When the intervention is finished, the outlet valve is opened and the pressure in the controlled wheel discharged to the reservoir. As soon as the driver releases the brake pedal, the fluid is pumped from the reservoir back to the brake-fluid reservoir.

Monitoring system

A comprehensive safety-monitoring system is of fundamental importance for reliable ESC functioning. The system used encompasses the complete system together with all components and all their functional interactions. The safety system is based on safety methods such as, for example, FMEA (Failure Mode and Effects Analysis), FTA (Fault Tree Analysis) and error-simulation studies. From these, measures are derived for avoiding errors which could have safety-related consequences. Extensive monitoring programs guarantee the reliable and punctual detection of all sensor errors which cannot be prevented completely. These programs are based on the well-proven safety software from the ABS and TCS systems which monitor all the components connected to the ECU together with their electrical connections, signals, and functions. The safety software was further improved by utilizing the possibilities offered by the additional sensors, and by adapting them to the special ESC components and functions.

The sensors are monitored at a number of stages. In the first stage, the sensors are continuously monitored during vehicle operation for line break, signal implausibility ("out-of-range" check), detection of interference, and physical plausibility. In a

second stage, the most important sensors are tested individually. The yaw-rate sensor is tested by intentionally detuning the sensor element and then evaluating the signal response. Even the acceleration sensor has internal background monitoring. When activated, the pressure-sensor signal must show a predefined characteristic, and the offset and amplification are compensated for internally. The steering-angle sensor has its own integrated monitoring functions and directly delivers a message to the ECU in the event of error. In addition, the digital signal transmission to the ECU is permanently monitored. In a third stage, analytical redundancy is applied to monitor the sensors during the steady-state operation of the vehicle. Here, a vehicle model is used to check that the relationships between the sensor signals, as determined by vehicle motion, are plausible. These models are also frequently applied to calculate and compensate for sensor offsets as long as they stay within the sensor specifications.

In case of error, the system is switched off either partially or completely depending on the type of error concerned. The system's response to errors also depends on whether the control is activated or not.

Further names for driving-dynamics control

Aside from ESP® and the neutral designation ESC for Electronic Stability Control there are further names used by vehicle manufacturers for driving-dynamics control. Examples are Dynamic Stability Control (DSC), Vehicle Stability Assist (VSA), Vehicle Stability Control (VSC), Dynamic Stability and Traction Control (DSTC), Controllo Stabilità e Trazione (CST).

Special driving-dynamics control system for commercial vehicles

Function

Heavy commercial vehicles essentially differ from passenger cars in their much greater mass combined with higher centers of gravity and in additional degrees of freedom resulting from trailer operation [7]. They can thus assume unstable states that extend far beyond the skidding familiar to passenger cars. Such states include, as well as the jackknifing of multiple-stage vehicle combinations, caused for example by trailer sliding, overturning caused by high lateral acceleration. A vehicle-dynamics control system for commercial vehicles must therefore, in addition to providing the stabilization functions familiar to passenger cars, also prevent jackknifing and overturning.

Requirements

The following requirements, in addition to those for passenger cars, can be derived from the extended functions of driving-dynamics control for commercial vehicles:

- Improvement in directional stability and response of a vehicle combination (e.g. articulated road train or articulated-train combination) at physical driving limits in all operating and laden states. This includes preventing jackknifing on vehicle combinations.
- Reduced risk of overturning for a vehicle or vehicle combination in both quasi-stationary and dynamic vehicle maneuvers.

These requirements implemented in commercial-vehicle ESC lead, as is the case with passenger cars, to a significant improvement in driving safety. For this reason, European law will from 2011 require the gradual introduction of a driving-dynamics control system for heavy commercial vehicles (from 7.5 t) (integral part of ECE-R 13 [11]).

Application

In the meantime, commercial-vehicle ESC has become available for virtually all vehicle configurations (except all-wheel-drive vehicles):

- vehicles with the wheel formulas 4×2 , 6×2 , 6×4 and 8×4 ,
- combinations of tractor unit and semitrailer (articulated vehicle or simply semitrailer unit),
- combinations of rig and drawbar trailer (articulated road train),
- multiple trailer combinations (Eurokombi), e.g. combinations of rig, dolly and semitrailer or semitrailer unit with additional center-axle trailer or tractor unit with B-link and semitrailer.

Operating principle

Driving-dynamics control for commercial vehicles can be divided according to requirements into the two function groups described in the following.

Stabilizing the vehicle in the event of imminent skidding or jackknifing

Directional stabilization of a commercial vehicle is initially performed according to the same principles as for a passenger car. The controller compares current vehicle motion with vehicle motion desired by the driver, taking into account physical driving limits. The physical model of motion in the horizontal plane – for a single vehicle characterized by three degrees of freedom (longitudinal, lateral and yaw motion) – is however extended for an articulated vehicle to include the articulation angle between the rig and the trailer (an additional degree of freedom). There are further degrees of freedom involved for combinations with fifthwheel trailers.

To calculate the vehicle motion desired by the driver, the ECU uses simplified mathematical-physical models (single-track vehicle model, [8]) to determine the nominal yaw velocity of the rig. The parameters that are encountered in these models (characteristic vehicle speed v_{ch} , wheelbase l and steering ratio i_L) are either parameterized at the end of the vehicle assembly line or adapted to the vehicle's behavior during vehicle operation with the aid of special adaption algorithms (e.g. Kalman filters or recursive least-

squares estimators, [9]). "Online" adaptation of parameters is particularly important in commercial vehicles, because the variety of variants and loads is much greater than for passenger cars.

Parallel to this, ESC calculates current vehicle motion from the measured variables available for yaw rate and lateral acceleration plus the wheel speeds. A significant deviation between the current vehicle motion and the motion expected by the driver leads to a control fault that is transformed by the actual controller into a corrective nominal yaw moment.

The level of the nominal yaw moment for a commercial vehicle depends on the control fault, the current vehicle configuration (wheelbase, number of axles, operation with or without trailer etc.) and the laden state (mass, center of gravity in the linear direction, etc.). As these parameters are variable, they are continuously determined by ESC. This is achieved, for example, in the laden state with the aid of an estimation algorithm that uses the signals from the engine management (engine speed and engine torque) and the vehicle linear motion (wheel speeds) to permanently identify the current vehicle mass.

On the basis of the current driving situation, the nominal yaw moment by braking individual or several wheels and of the trailer is transformed in a suitable manner. This is depicted by way of example in Figure 11a and in Figure 11b for a clearly defined oversteer and understeer situations respectively.

In addition to these clearly defined situations, there are other critical dynamic situations in which other wheels or wheel combinations are braked depending on the desired stabilization effect. Thus, for example in the case of sharper understeer, the entire vehicle is braked along similar lines to Enhanced Understeering Control (EUC) in passenger cars.

Because of the high center of gravity of commercial vehicles, skidding and jackknifing by such vehicles primarily occur at low-to-medium coefficients of friction at which the tire static-friction limit is already exceeded at an early stage. At high coefficients of friction, laden commercial vehicles, on account of their high center

of gravity, normally start to overturn before the static-friction limit of the tires is reached.

Reducing the risk of overturning

The overturning limit (lateral-acceleration limit) of a vehicle depends not only on the height of the center of gravity but also on the chassis systems (axle suspension, stabilizers, springs, etc.) and the type of payload (fixed or moving) [10].

The situation which causes a commercial vehicle to overturn is, aside from a relatively low overturning limit, an excessive cornering speed. ESC makes use of this scenario to reduce the probability of the vehicle overturning. As soon as the vehicle approaches the overturning limit, it is slowed down by reducing engine torque and, if necessary, also applying the brakes. The overturning limit is determined here depending on the load of the vehicle and the load distribution, whereby the laden state of the vehicle is estimated "online".

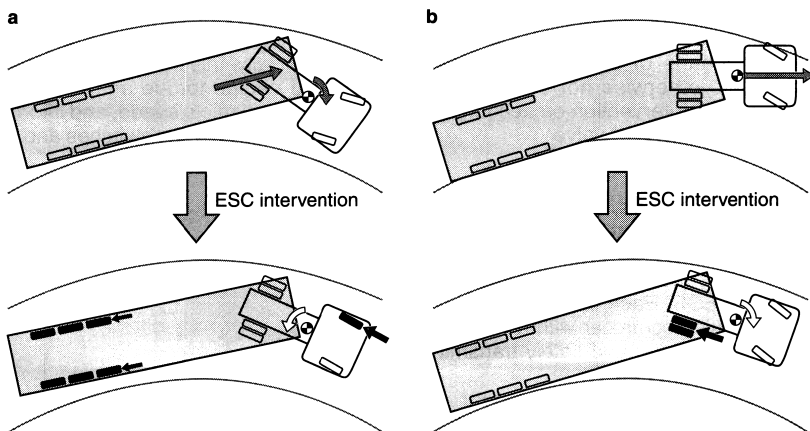
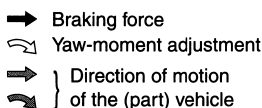
Depending on each driving situation, the overturning limit that is determined is modified. Thus, the overturning limit in high-speed dynamic situations (e.g. obstacle-avoidance maneuvers) is reduced in order to permit early intervention. In very slow maneuvers, on the other hand (e.g. negotiating tight hairpin bends on uphill stretches), it is increased in order to prevent unnecessary and disruptive ESC intervention.

Determining the overturning limit is based on various assumptions regarding the height of the center of gravity and the dynamic response of the vehicle combination with a known axle-load distribution. This covers the largest portion of the usual vehicle combinations.

In order to ensure stabilization even in the case of strong deviations from these assumptions (e.g. extremely high centers of gravity), ESC additionally detects the lift of the wheels on the inside of a bend. This is achieved by monitoring the wheels

Figure 11: Principle of ESC braking intervention on semitrailers

- a) Vehicle oversteering,
b) Vehicle understeering.



for implausible rotation speed. If necessary, the entire vehicle combination is then heavily decelerated by brake intervention.

A trailer wheel lifting on the inside of a bend is indicated by the trailer's electronically controlled braking system (ELB) via the CAN communication line (ISO 11992 [13]) by activating the ABS controller. For combinations with trailers equipped with ABS only, wheel-lift detection on the inside of a bend is limited to the tractor unit (rig).

System design

On the European market, the electronically controlled braking system ELB has come to the fore as the standard for brake control in heavy commercial vehicles. ESC is based on this system, extending it to include regulation of the driving dynamics. To do so, ESC uses the ELB capability of generating varying braking forces for each individual wheel independently of driver action.

The very different general conditions for commercial vehicle brake systems in North America mean that purely ABS or ABS/TCS systems are used as standard. An ESC based on ABS/TCS is therefore used for these and similar markets. Here, ESC uses the method already applied with TCS on the drive axle to generate braking force individually for each wheel independently of the driver by means of a TCS valve and the downstream ABS valves. In addition, for ABS-based ESC, the driver brake command must be measured by means of pressure sensors, which would otherwise not be possible during an ESC intervention on account of the function of the TCS valve.

Sensor systems

Like passenger cars, commercial vehicles use a combined yaw-rate and lateral-acceleration sensor and a steering-wheel-angle sensor as driving-dynamics sensors for ESC. Each of these sensors contains a microcontroller with a CAN interface for analyzing and safely transmitting the measured data.

The steering-wheel-angle sensor is usually mounted immediately below the steering wheel and it measures the angle

of rotation of the steering wheel. This is then converted in the ECU into a wheel steering angle.

In order to pick up the lateral acceleration as close as possible to the center of gravity of the rig, the combined yaw-rate and lateral-acceleration sensor is usually mounted in the vicinity of the center of gravity.

Even though commercial vehicles essentially use the same sensors as passenger cars, the yaw-rate and lateral-acceleration sensor must have a much more robust design to cope with the rougher ambient conditions, particularly on the commercial-vehicle frame.

Electronic control unit (ECU)

The ESC algorithms are run together with the other algorithms for brake control (e.g. ABS and TCS) in the brake control unit. This control unit is constructed using conventional circuit-board technology with correspondingly powerful micro-controllers.

A CAN bus connects the ESC sensors with the control unit. The nominal brake pressures and wheel slip values of the ESC are then implemented by the relevant braking system for each wheel and for the trailer. In addition, the braking system transmits the requested engine torque via the vehicle CAN bus (usually standardized as per SAE J 1939 [14]) to the engine ECU for implementation.

Moreover, relevant information is also transferred from the engine and retarder to the braking system via the vehicle CAN bus. Essentially, this involves current and requested engine torque and speed, retarder torque, vehicle speed, and information from various control switches and any trailer that may be coupled.

Safety and monitoring functions

The extensive possibilities for ESC intervention in the handling characteristics of the vehicle and vehicle combination require a comprehensive safety system to ensure proper system functioning. This extends not only to the basic ELB or ABS/TCS system respectively, but also to the additional ESC components, including all sensors, ECUs and interfaces.

The monitoring functions used for ESC are essentially based on the functions used in passenger cars and are adapted to the characteristics of commercial vehicles.

There is also mutual monitoring of the microcontrollers distributed in the overall system. This means that the brake control unit contains a main computer and a monitoring computer that mainly performs plausibility checks alongside minor functional tasks. Furthermore, the corresponding algorithms permanently check memory and other internal computer hardware components to detect any defects that occur at an early stage.

The occurrence of faults results, depending on the nature and significance of the fault, in the shutdown of individual function groups through to a complete switching to "backup mode", in which the brakes are controlled by purely pneumatic means (fail-silent response). This ensures that incorrect sensor signals cannot cause implausible and possibly dangerous operating states in braking operations.

The occurrence of a fault is indicated to the driver by suitable means (e.g. warning lamp or display) so that suitable action can be taken.

Furthermore, any faults that occur are assigned a time stamp in the control unit and stored in the fault memory. The workshop can evaluate these with the help of a suitable diagnosis system.

References

- [1] E.K. Liebermann, K. Meder, J. Schuh, G. Nenninger: Safety and Performance Enhancement: The Bosch Electronic Stability Control. SAE Paper Number 2004-21-0060.
- [2] National Highway Traffic Safety Administration (NHTSA) FMVSS 126: Federal Motor Vehicle Safety Standards; Electronic Stability Control Systems; Controls and Display. Vol. 72, No. 66, April 6, 2007.
- [3] M. Mitschke, H. Wallentowitz: Dynamik der Kraftfahrzeuge. 4th Edition, Springer-Verlag, 2004.
- [4] A. van Zanten, R. Erhardt, G. Pfaff: FDR – Die Fahrdynamikregelung von Bosch. ATZ Automobiltechnische Zeitschrift 96 (1994), Volume 11.
- [5] A. Trächtler: Integrierte Fahrdynamikregelung mit ESP®, aktiver Lenkung und aktivem Fahrwerk. at – Automatisierungstechnik 53 (1/2005).
- [6] K. Reif: Automobilelektronik, 3rd Edition, Vieweg+Teubner, 2009.
- [7] E. Hoepke, S. Breuer (Editors): Nutzfahrzeugtechnik – Grundlagen, Systeme, Komponenten. 4th Edition; Vieweg Verlag, 2006.
- [8] C.B. Winkler: Simplified Analysis of the Steady State Turning of Complex Vehicles. International Journal of Vehicle Mechanics and Mobility, 1996.
- [9] Ali H. Sayed: Adaptive Filters. John Wiley & Sons, 2008.
- [10] D. Odenthal: Ein robustes Fahrdynamik-Regelungskonzept für die Kippvermeidung von Kraftfahrzeugen. Dissertation TU München, 2002.
- [11] ECE-R13: Uniform provisions concerning the approval of category M, N and O vehicles with regard to brakes.
- [12] ECE-R13-H: Standard conditions for approval of passenger cars with regard to brakes.
- [13] ISO 11992: Road vehicles – Interchange of digital information on electrical connections between towing and towed vehicles.
- [14] SAE J 1939: Serial Control and Communications Heavy Duty Vehicle Network Top Level Document.

Automatic brake-system operations

The supplementary functions described here utilize except for the pneumatic-mechanical brake assistant the driving-dynamics control infrastructure. This comprises the sensors, the actuators and the ECU.

Brake-assistant operations

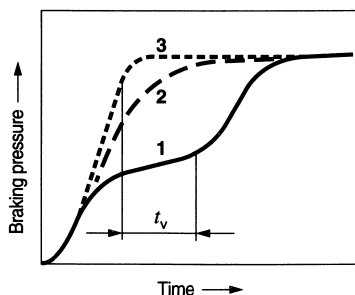
Investigations into driver braking behavior in the 1990s showed that car drivers differ in the way that they respond to braking situations. The majority – the “average drivers” – do not brake hard enough when faced with an emergency situation, in other words, they require an unnecessarily long braking distance (Figure 1). This can be remedied by a system first launched in 1995, the brake assistant. Its main purposes are as follows:

- It interprets a certain rate of pedal movement (rapid application of the brakes) that fails to apply maximum braking force as an intention by the driver to carry out full braking. In this case, it generates the braking pressure required to achieve full braking effect.
- It allows the driver to “cancel” full braking operation at any time.

Figure 1: Comparison between braking with and without brake-assistant operations

A longer braking distance is required without a brake assistant than with a brake assistant.

- 1 “Average driver”,
 - 2 Experienced driver,
 - 3 “Average driver” with brake assistant.
- t_v Delay time for braking results in longer braking distance.



UFB0709-3E

- The behavior of the brake booster and, therefore, the pedal feedback, is not altered under normal braking conditions.
- The basic braking-system function is not diminished if the brake assistant fails.
- The system is designed to prevent accidental activation.

Pneumatic brake assistant

This system requires a modified brake booster which increases the level of amplification according to the rate of pedal movement and pedal force. This results in faster and greater buildup of pressure in the wheel brakes.

In an alternative version, the brake booster is extended to include an electronically activated valve. This enables an ECU to influence the pressure difference between the brake-booster chambers and thus the boosting of the braking force. This provides better opportunities for optimizing the trigger threshold and response characteristics.

Hydraulic brake assistant

The hydraulic brake-assistant operation uses the driving-dynamics control hardware. A pressure sensor detects the driver's braking intention; the ECU analyzes the signal on the basis of the defined triggering criteria, and initiates an appropriate brake-pressure build-up in the hydraulic system. The upstream brake booster is a standard unit and does not require any modification.

As a general observation, it should be stated that it is an absolute requirement that all brake-assistant system variants referred to are used in conjunction with an antilock braking system (ABS) or a driving-dynamics control system due to the actively generated rapid brake-pressure rise beyond the wheel-lock limit.

Automatic brake-pressure increase at the rear wheels

This is a function which provides the driver with additional brake servo assistance for the rear wheels if the front wheels are controlled by the ABS system. This function was introduced because many drivers do not increase the pedal force at the start of ABS control even though the situation would require this. Once ABS control has been initiated on the front wheels, the wheel pressures at the wheels on the rear axle are increased by way of the return pump of the hydraulic modulator until these too have reached the lockup-pressure level and ABS control is initiated (Figure 2). The brake application is therefore at the physical optimum. The pressure in the rear axle wheel-brake cylinders can then exceed the pressure in the brake master cylinder, also during ABS control.

The cutoff condition is fulfilled when the wheels on the front axle are no longer under ABS control or when the pressure in the brake master cylinder falls below the cutoff threshold.

Automatic brake-pressure increase with forceful pressing of brake pedal

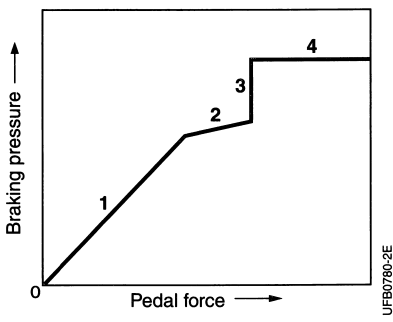
This function provides the driver with additional brake servo assistance. It is activated if the maximum possible vehicle deceleration is not achieved even if the driver forcefully presses the brake pedal to the point that would normally cause the lockup pressure to be reached (primary pressure over approx. 80 bar). This is the case, for example, at high brake-disk temperatures or if the brake pads have a considerably reduced coefficient of friction.

When this function is activated, the wheel pressures are increased until all wheels have reached the lockup pressure level and ABS control is initiated (Figure 2). The brake application is therefore at the physical optimum. The pressure in the wheel-brake cylinders can then exceed the pressure in the brake master cylinder, also during ABS control.

If the driver reduces the desired level of braking to a value below a particular threshold value, the vehicle deceleration is reduced in accordance with the force applied to the brake pedal. The driver can therefore precisely modulate the vehicle deceleration when the braking situation has passed. The cutoff condition is fulfilled if the primary pressure or vehicle speed falls below the respective cutoff threshold.

Figure 2: Automatic brake-pressure increase

- 1 Brake-pressure increase by brake booster,
- 2 Further brake-force increase by pedal force,
- 3 Brake-pressure increase by driving-dynamics control hydraulic modulator,
- 4 ABS control range.



Brake-disk wiper

This function ensures that, in the event of rain or a wet road, the splash water is cyclically removed from the brake disks. This is achieved by automatically setting a low brake pressure at the wheel brakes. In this way, the function helps to ensure minimum brake-response times when driving in wet conditions. Evaluated signals from the windshield wipers or rain sensor are used to detect wet conditions.

The pressure level is adjusted so that the vehicle deceleration cannot be perceived by the driver. Actuation is repeated at a defined interval for as long as the system detects rain or a wet road. If required, just the disks at the front axle can be wiped. The wiping procedure is terminated as soon as the driver applies the brakes.

Automatic prefill

This function reduces the total braking distance in emergency situations whereby the driver operates the brake pedal immediately after the accelerator pedal is released. This is achieved by prefilling and thereby preloading the brake system after release of the accelerator pedal, which means that in the subsequent brake operation the pressure build-up is considerably more dynamic. Accordingly, high vehicle deceleration sets in earlier.

Prefilling of the brake system is adjusted by the return pump of the hydraulic modulator. The brake shoes are then firmly applied to the brake disks. If there is no operation of the brake pedal directly after a rapid release of the accelerator, the pressure in the brake system is reduced again. This does not impair driveability.

Electromechanical parking brake

The electromechanical parking brake (EMP) generates the force for applying the parking brake by electromechanical means. The function of the hand- or foot-operated parking-brake lever is performed by a control knob with an electric-motor-and-transmission combination. When the driver operates the control knob, the electric motor (actuator) is activated when the system detects that the vehicle is at a standstill. When the vehicle is parked on a level surface, the holding forces are set lower than when the vehicle is fully laden and parked on a gradient. Active wheel sensors are used to detect that the vehicle is parked (i.e. stationary). As an option, it is also possible for the road gradient to be detected by a tilt sensor.

The parking brake is released by means of the same control knob. However, various safety regulations and requirements have to be met, e.g. to prevent inappropriate or inadvertent release of the parking brake by children or animals (see also Parking brake, Passenger-car brake system).

Controlled braking with hydraulic modulator

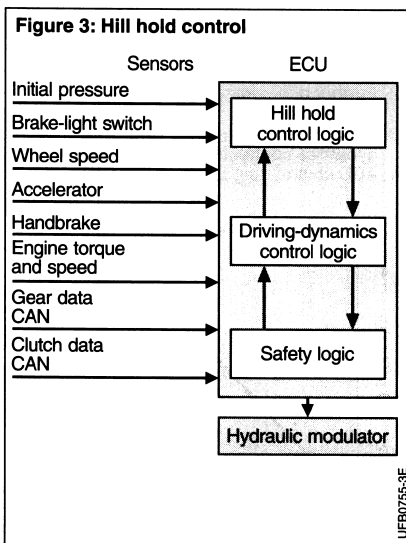
If the electromechanical parking brake is activated while driving, the vehicle must be safely decelerated to a stop. The brake pressure required for this is built up by the return pump of the hydraulic modulator. The driving-dynamics control system ensures safe braking even on slippery or wet road surfaces. When the vehicle has come to a stop, the electromechanical parking brake assumes the function of keeping the vehicle stationary.

The driver must press the activation knob of the automatic parking brake continuously during the deceleration phase.

Hill hold control

This system simplifies hill starts. It prevents the vehicle from rolling back after the driver has released the brake pedal. This is particularly helpful on heavily laden vehicles with manual transmission and vehicles that are towing trailers. There is no need to operate the parking brake. The function also works when performing a hill start in the reverse direction.

The system detects the driver's intention to pull away (Figure 3). After the brake pedal is released, there is approximately two seconds to start pulling away. The



brake is released automatically here if the drive torque is greater than the torque resulting from downgrade force.

Low-speed traction control is based on the driving-dynamics control hardware and additional sensors: a tilt sensor detects the road gradient, a gear switch detects whether the driver has shifted to reverse gear, and a clutch switch recognizes whether the driver has depressed the clutch pedal.

Automatic braking on hill descent – hill descent control

This system is a convenience function which assists the driver on offroad descents with gradients of approximately 8 to 50% by automatically operating the brakes. The driver can then concentrate fully on steering the vehicle and is not distracted by the need to operate the brakes at the same time. The brake pedal does not have to be operated.

When this system is activated, e.g. by pressing a button or switch, a preset speed is maintained over the extent of the specified brake pressure. If required, the driver can vary the predetermined speed by pressing the brake and accelerator pedal or using the control buttons of a speed control system.

The system remains active until it is switched off by pressing the button or switch again, i.e. it is not automatically deactivated.

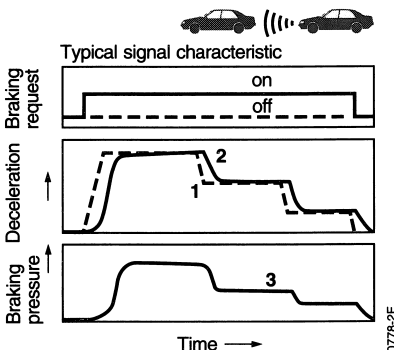
Automatic brake application for driver assistance systems

This function is an additional function for active brake application with adaptive cruise control (ACC), i.e. for automatic vehicle-to-vehicle ranging. The brakes are applied automatically without the driver pressing the brake pedal as soon as the distance to the vehicle in front falls below a predetermined distance (Figure 4). This is based on a hydraulic brake system and driving-dynamics control.

The function receives a request to decelerate the vehicle by a desired amount (input). This is calculated by the upstream ACC system. Automatic brake intervention maintains vehicle deceleration by means of appropriate brake pressures which are adjusted with the aid of the hydraulic modulator.

Figure 4: Function description of automatic brake application for ACC

- 1 Deceleration specified by ACC (Adaptive Cruise Control),
- 2 Current deceleration,
- 3 Pressure in wheel-brake cylinders.



Integrated driving-dynamics control systems

Overview

In addition to the brakes and engine, as used by the driving-dynamics control system (Electronic Stability Control, ESC), there are other actuators in the chassis and in the drivetrain which are used to purposefully influence driving dynamics. The functional combination of actuators relevant to driving dynamics is referred to by different names by the various automobile manufacturers and suppliers: Vehicle Dynamics Management (VDM), Integrated Chassis Control (ICC), Integrated Chassis Management (ICM), and Global Chassis Control (GCC).

Primary uses

The functions are used primarily in the fields of directional stability, agility and reducing "driver workload", i.e. the efforts exerted by the driver to operate and control the vehicle.

Improving directional stability

In critical driving situations, these functions take on stabilization tasks which are normally performed by the driver himself. Typical examples are the oversteer situation or braking on roads with different grip factors on the left and right sides of the vehicle (μ split).

Increasing agility

Some of these functions improve the acceleration response of the lateral-dynamic response of the vehicle to steering inputs by the driver. These make the vehicle more agile, i.e. it responds more spontaneously and dynamically to driver inputs.

Reducing the effort to control/operate the vehicle

An improved response by the vehicle to movements of the steering wheel and automatic stabilization interventions relieve the driver's workload, the steering effort in particular being reduced.

Functions

Driving-dynamic driver steering recommendation

Function

This function utilizes electrical power steering as the actuator. An additional, driving-dynamically motivated steering torque is superimposed on the steering-servo torque of the power steering in order to provide the driver with a steering recommendation.

Limitation of supplementary steering torque

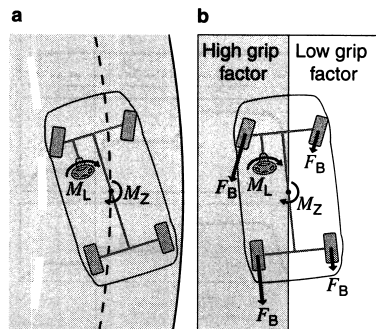
The driving-dynamic supplementary steering torque is limited, depending on the vehicle, to values of approximately 3 Nm so that the driver can oversteer the steering recommendation at any time.

Driving-dynamics applications

The steering recommendation is activated in different driving situations. In a oversteer situation, a steering torque is introduced in the countersteer direction (Figure 1a). In the event of understeer, the function motivates the driver not to increase the steer

Figure 1: Driving-dynamic driver steering recommendation with supplementary steering torque

- a) Oversteer,
 - b) Braking on road with different grip factors.
- M_L Supplementary steering torque,
 M_Z Yaw moment based on intended driver reaction,
 F_B Braking force.



angle further, since the side-force potential on the front wheels is already exhausted and further steering may even reduce the side force.

When braking or accelerating on roads with different grip factors on the left and right sides of the vehicle (μ split), the driver is assisted in compensating the vehicle yaw motion by countersteering (Figure 1b).

Active steering stabilization on the front axle

Function

This function intervenes via the angle-superimposing steering actuator of the override steering directly in the vehicle movement. An additional steering angle adjusted by an electric motor is added to the driver's steering angle in an override gearbox.

Driving-dynamics applications

The function operates, for example, in an oversteering driving situation. Yaw-rate control with automatic steering-angle intervention returns the excessively high yaw-rate value to its setpoint value.

During braking on roads with different grip factors, the function provides for compensation of the yaw motion. Since yaw compensation is automatically triggered and happens much faster than the steering operation of a typical driver, the measures in driving-dynamics control for yaw-moment build-up delay can be reduced. The braking distance is also reduced.

A further benefit can be achieved in the event of understeer and starting on roads with different grip factors.

Demands on the sensor system

Because of the high positioning rate of an override steering system, the sensor signals must be monitored with low fault latencies, i.e. a fault must be signaled quickly. This means that the inertial sensor system must have a redundant design.

Active steering stabilization on the rear axle

Function

In vehicles with rear-axle steering, the primary function controls the rear-axle steering angle as a function of the steering-wheel angle and the driving speed. At low speeds, the rear wheels are steered in the opposite direction to the front wheels, which improves vehicle maneuverability.

At high speeds, the rear wheels are steered in the same direction as the front wheels. In this way, the yaw motion is less intensely excited during obstacle-avoidance maneuvers. This increases directional stability considerably.

Driving-dynamics applications

The associated functions act in situations similar to active front-axle steering interventions. The effectiveness of oversteer interventions is crucially dependent on the current utilization of the side-force potential on the rear wheels.

When braking on roads with different grip factors, it is necessary on account of the reduction of load on the rear axle to make interventions with larger angles in order to suppress the yaw motion. On the other hand, braking can take place without a considerable attitude angle.

Demands on the sensor system

The functions for rear-axle steering likewise require a redundant inertial sensor system, since the positioning rate of a steering actuator on the rear axle is comparable with the override steering.

Torque distribution in the longitudinal direction for four-wheel drives

Function

In vehicles with four-wheel drive, interaxle differential locks or transfer cases within the framework of their basic function improve the traction and, with it, the accelerating performance of the vehicle, whereby the wheels of both axles transfer tractive forces.

Driving-dynamics applications

In situations where the grip factor is not fully utilized by traction, the self-steering properties can be altered by shifting the tractive forces between the front and rear axles. The more tractive force an axle transfer, the more the side-force potential on the wheels of this axle is weakened. Shifting the tractive force to the front axle increases the understeer tendency, while shifting it to the rear axle reduces the understeer tendency. Through dynamic control of drivetorque distribution, lateral-dynamic agility can be increased without the vehicle tending towards oversteer.

Wheel-torque distribution in the lateral direction

Function

Functions based on a "torque-vectoring actuator" make a significant contribution to increasing agility. The actuator permits an extensively free shift of the wheel torque between the left and right wheels on an axle. In this way, the total turning force excluding friction losses is not reduced in the actuator meaning that this intervention is virtually neutral with regard to the driving speed.

Driving-dynamics applications

In an understeering driving situation, the drive torque of the wheel on the outside of the curve is increased and the drive torque of the wheel on the inside of the curve is decreased. In this way, an additional yaw moment turning into the curve acts on the vehicle (Figure 2a). The understeer tendency decreases; the vehicle's lateral-dynamic agility increases.

When accelerating on roads with different grip factors, the drive torque is specifically directed to the wheel with a high grip factor. Brake interventions by the traction control system on the wheel with a low grip factor are dispensed with to a large extent. This increases the mean acceleration on roads with different grip factors.

Even in an oversteering situation, the brake interventions are partly replaced by a wheel-torque shift (Figure 2b). In this way, the speed loss can be reduced by brake interventions of the driving-dynamics control in the event of mild oversteer. In critical oversteering situations, the brakes still intervene, since in this case the speed loss is desired to alleviate the driving situation.

Replacement of wheel-torque distribution by brake and engine-control interventions

The understeer interventions by means of a torquevectoring actuator can be emulated by brake interventions on the wheels on the inside of the curve. The engine torque is increased to compensate the deceleration by the one-sided brake intervention. This produces an effect similar to understeer intervention with a torque-vectoring actuator. This only requires a hydraulic modulator with a long service life and low noise generation instead of an additional torque-vectoring actuator.

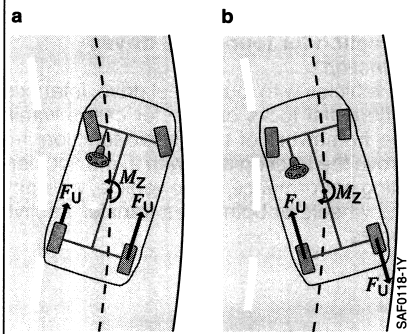
Influencing self-steering properties through roll stabilization

Function

In line with their comfort-oriented basic function, systems for roll stabilization serve to compensate the rolling motion of the vehicle body when cornering.

Figure 2: Wheel-torque distribution in the lateral direction

- a) Understeer,
- b) Oversteer.
- F_U Wheel peripheral force (driving, braking),
- M_Z Yaw moment based on intervention.



Driving-dynamics applications

Where the roll-stabilization system has a two-channel design with separate activation of the actuators on the front and rear axles, there is an additional option for influencing the self-steering properties. The roll-compensating torques can – within the framework of the performance of the individual channels – be actively distributed between the front and rear axles. This results in a change of the wheel downforces.

The effect on the self-steering properties is based on the degenerative increase in the lateral forces with the normal forces (Figure 3). When the rolling moment is extensively supported over the rear axle, the normal force at the rear on the wheel on the outside of the curve increases dramatically; the associated side force, however, increases only underproportionally. The side force on the rear axle is weakened, and the understeer tendency decreases. As a countermove, the understeer tendency increases when the rolling moment is extensively supported over the front axle.

Influencing self-steering properties through variable vibration dampers*Function*

The use of variable dampers for adaptive damping of the vertical movement of the vehicle body has become significantly widespread. Pitching and rolling motions which are caused by driver inputs such as braking and steering or by road-surface defects, as well as vertical movements caused by road-surface defects, can be noticeably reduced.

Driving-dynamics applications

It is also possible with limited effect to influence the self-steering properties. The action mechanism corresponds to influencing the self-steering properties through roll stabilization. However, the function only has a transient effect during a rolling motion of the vehicle body, since a damper movement is a prerequisite for damping forces. The damping forces can be modulated by adjusting the damping hardness so that the wheel downforces can be semi-actively distributed as described above.

System architecture**Allocation of functions to ECUs**

The actuators in the chassis and in the drivetrain are generally activated not only by functions of the integrated driving-dynamics control systems, but also first and foremost by basic functions which are very closely connected with the actuator. Examples of such basic functions include servo assist in an electrical power-steering system or variable steering ratio in an override steering system.

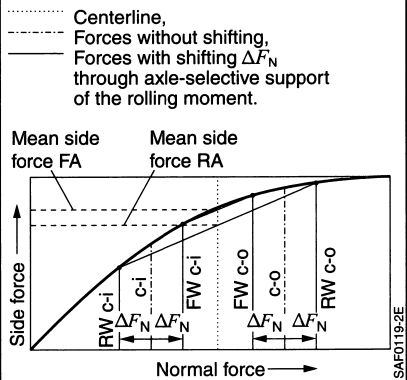
The basic functions are characterized by moderate networking with other vehicle systems. Because of their close connection to the actuator, they are usually integrated into the associated ECU, which also activates and monitors the actuator.

On the other hand, the functions of the integrated driving-dynamics control systems are highly networked, in particular with driving-dynamics control (ESC) and other chassis control systems. They calculate a setpoint driving-dynamic value for the actuator, which is transmitted via a data bus to the ECU. An arbitration with the setpoint values of the basic functions is performed in the ECU.

Figure 3: Side force as a function of normal force with a constant slip angle

Reduction of understeer tendency by shifting of normal wheel forces.

FA Front axle, RA Rear axle,
FW Front wheel, RW Rear wheel,
c-i Inside of curve, c-o Outside of curve.



The ECU for driving-dynamics control (ESC) or a central ECU of the "Chassis" functional area is suitable for use as the integration platform for the functions of the integrated driving-dynamics control systems (Figure 4).

Interaction of various functions

Various functions and actuators are increasingly being installed in a single vehicle. This calls into question a function structure which guarantees good interaction of the individual functions and which above all prevents mutual interference.

The original trend observed of implementing a separate controller for each actuator is reaching its limits. However, fully bringing together all the algorithms in a central controller limits flexibility in the distributed development and in the use of ECU resources.

A more promising compromise is bringing together all the controllers which utilize a related positioning principle, since interventions with a related positioning principle require particularly intensive co-ordination.

The following are mentioned as positioning principles and allocated positioning systems:

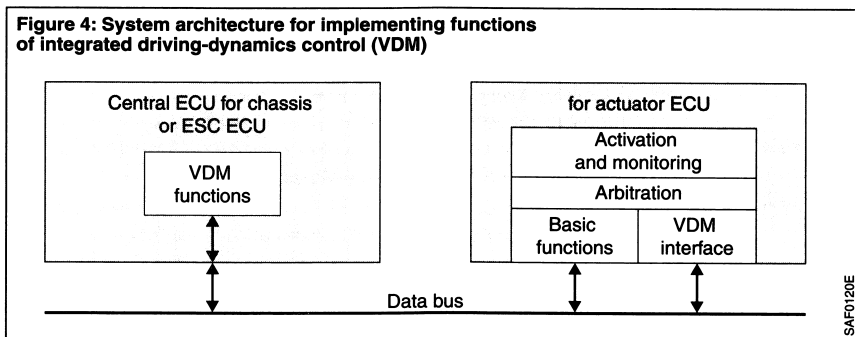
- wheel torque: driving-dynamics control, controllable differential locks, torque vectoring,
- steering angle: override steering (front axle) and rear-axle steering,
- normal force: roll stabilization, adjustable shock absorbers.

The driving-dynamic steering-torque recommendation plays a special role. It must be matched in terms of its effect on the driver with functions for override steering.

References

- [1] Isermann (Editor): *Fahrdynamikregelung – Modellbildung, Fahrerassistenzsysteme*, Mechatronik. Vieweg Verlag, Wiesbaden, 2006.
- [2] Deiss, Knoop, Krimmel, Liebermann, Schröder: *Zusammenwirken aktiver Fahrwerk und Triebstrangsysteme zur Verbesserung der Fahrdynamik*. 15th Aachen Colloquium Automotive and Engine Technology, Aachen 2006, PP. 1671...1682.
- [3] Klier, Kieren, Schröder: *Integrated Safety Concept and Design of a Vehicle Dynamics Management System*. SAE Paper 07AC-55, 2007.
- [4] Erban, Knoop, Flehmig: *Dynamic Wheel Torque Control – DWT*. Agility enhancement by networking of ESC and torque vectoring. 8th European All-wheel Congress, Graz 2007, PP. 17.1...17.10.
- [5] Flehmig, Hauler, Knoop, Münkel: *Improvement of Vehicle Dynamics by Networking of ESC with Active Steering and Torque Vectoring*. 8th Stuttgart International Symposium Automotive and Engine Technology, Report Vol. No.2, PP.277...291, Vieweg Verlag, Wiesbaden 2008.

Figure 4: System architecture for implementing functions of integrated driving-dynamics control (VDM)



Passive safety

There are 150 milliseconds which decide in the event of a car accident whether the outcome remains confined to damage to property or escalates to serious injuries to the occupants. If an accident occurs, numerous vehicle components can be crucial to survival. From the passenger safety cell extending across a stable body, crumple zone, and headrests through to airbags and seat belts with belt pretensioners (see Occupant-protection systems), passive safety is aimed at mitigating the consequences of an accident.

Whereas active safety aims at avoiding an impending accident entirely, passive safety is intended to mitigate the consequences of an unavoidable accident. By linking with driver assistance and active safety, new enhanced passive-safety functions provide for more comprehensive occupant protection (Figure 1).

Phases of vehicle safety

Vehicle safety is divided into five phases (Figure 1). Active safety includes normal driving, in which safety systems issue warnings, provide recommendations, or make corrections (phase 1 and phase 2). In the event of a collision passive vehicle safety intervenes in a protecting (phase 3 and phase 4) and in the post-crash phase in a life-saving capacity (phase 5).

Driving mode

Belt-reminder system

If the driver or front passenger does not have their seat belt fastened, there is a visual and acoustic warning signal (SBR, Seat Belt Reminder, belt-reminder system, Figure 2a). Sensing is performed for example by means of a seat-occupancy mat (see Occupant sensing). The presence of further occupants in the rear passenger compartment is optionally detected via sensors. In this case, the term used is Enhanced Seat Belt Reminder (ESR).

Figure 1: Vehicle-safety phases

ACC: Adaptive Cruise Control
PCW: Predictive Collision Warning
MSB: Motorized Seat Belt (electromechanical belt pretensioner)
PS-P: Presafe-Pulse, triggering before t_0

Brake pre-fill: Application of brake pads
AEB: Automatic Emergency Braking
SCM: Secondary Collision Mitigation
eCall: Emergency call

Phases	1	2	3	4	5
Situation	Normal driving	Critical situation	Accident avoidable	Accident	Post-crash
Action	Information	Preparation	Warning	Intervention	Protection
Timing	$t = -2000 \dots -1000 \text{ ms}$		$t = -600 \dots -200 \text{ ms}$	$t_0 = 0 \text{ ms}$	$t = 200 \text{ ms}$
Accident phase	Pre-crash			In-crash	Post-crash
Area	Driver assistant	Active safety		Passive safety	
	"New" Passive safety = Integrated safety (before, during and after t_0)				
Safety objective	Driver support	Accident avoidance		Reduction of accident consequences	
Example	ACC	Brake pre-fill	PCW	AEB	MSB PS-P Airbag SCM eCall

SKT0064-2E

Child seats

The use of child seats is required by law. Provisions also stipulate that the front-passenger airbag must be deactivated if a rear-facing child seat is placed on the front passenger seat (COP, Child Occupant Protection). The airbag can be manually or automatically switched depending on regional variation. The corresponding function lamp lights up when the airbag is deactivated (Figure 2b).

Before the crash

For improvement of the deployment function of the occupant-protection systems and advance detection of the type of impact (pre-crash detection), microwave radar, ultrasound or lidar sensors (optical system using laser light, see Sensors) are used to measure relative speed, distance, and angle of impact for frontal impacts. If distances are too short at a corresponding speed, optical and acoustic warnings can be issued or the brake can be activated (AEB, Automatic Emergency Brake, see Emergency brake systems). Electromechanical reversible seat-belt pretensioners are used with pre-crash sensing. The fact that they are reversible means that in a dangerous situation they can already be tightened prior to a possible collision. In this way, belt slack is already eliminated before the crash, enabling the occupants already at an early stage to participate in the vehicle's deceleration during a braking operation.

During the crash

The crash starts at t_0 , the moment of first vehicle contact (in-crash) with the obstacle, and ends with the vehicle at a standstill. In this time window all the rele-

vant retention systems are activated (see Occupant-protection systems) and if necessary emergency braking is initiated; this is intended to avoid a potential secondary accident or to mitigate the accident severity.

After the crash

Event Data Recorder

Vehicles with airbag systems store data relevant to the accident, e.g., impact accelerations, belt-buckle states or airbag deployment times, in a non-volatile memory of the airbag ECU (EDR, Event Data Recorder). The volume of data varies depending on the vehicle manufacturer and records a period of around 100 ms before and after the impact.

This enables the accident to be analyzed to a certain extent. For example, accidents with airbag deployment can be distinguished from near-accidents (near misses) and also accidents without airbag deployment. For another thing, the data permit a partial reconstruction of the vehicle situation before the accident for accident-research purposes. In addition, statutory requirements are thereby fulfilled. Thus, in the USA the use of an event data recorder in new vehicles has been required by law since 2012.

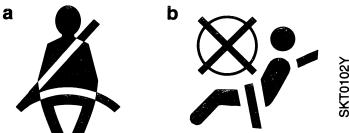
eCall

eCall (abbreviated form for emergency call) is an automatic emergency-call system for motor vehicles planned by the European Union which from April 2018 must be installed in all new models of passenger cars and light commercial vehicles.

The airbag ECU activates, aside from hazard warning lights, door openers, deactivation of the fuel pump, and isolation of the battery, an emergency call via a carphone installed in the vehicle. The emergency-call center automatically receives information on the exact location of the vehicle and can if necessary talk to the occupants and initiate recovery and rescue operations.

Figure 2: Warning lamps in the instrument cluster

- a) Symbol lights up if seat belt is not fastened.
- b) Front-passenger airbag deactivated.



**System architecture,
passive safety**

The passive-safety functions are specific to the manufacturer, vehicle, and equipment specification. Figure 3 shows a possible architecture.

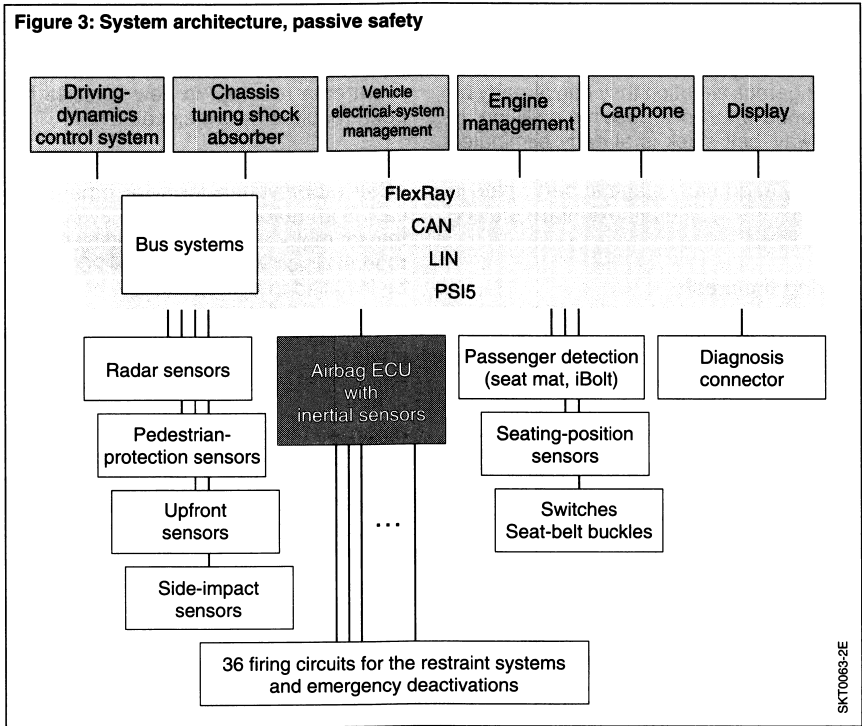
The airbag ECU collects information on the speed, temperature, belt-buckle states, the seating position of the occupants, and registers activation of the ignition. Diagnoses, driving states, and accidents are recorded in the vehicle and stored in an event recorder. Data can be read via the diagnosis connector and if necessary software updates can be performed.

Inertial-sensor signals, i.e., accelerations and yaw rates in all three directions, are also made available for other systems. In this way, spring-damping systems or even brake and steering systems can be activated with these signals to optimize and stabilize the vehicle in terms of driving dynamics.

The information already in the lead-up to a crash – e.g., on the relative speed, moment of impact of potential collision objects – comes from a radar, lidar or video sensor system. This enables seats or reversible belt pretensioners for example to be preconditioned for the crash situation or warning signals to be issued.

The vehicle electrical-system management uses the signals from the airbag ECU to control the airbag warning lamps, the status of the seat belts and seat occupancy, the hazard-warning lights, and emergency door opening.

An emergency call is sent via the carphone in the event of a crash; there is also the possibility of communicating information on medical treatment to the emergency services in advance. The engine management switches off the fuel pump after a crash; if necessary, the battery is isolated.



Legal regulations and consumer tests

Legal regulations

Requirements with regard to passive safety in vehicles are laid down in legislation. These cover safety of drivers, passengers, and in particular children and pedestrians.

Safety requirements laid down in the EU are for example the provisions pertaining to head-on- and side-impact protection for the vehicle occupants and pedestrian protection. These are reflected in UN/ECE regulations, which apply to the EU but have in the meantime also been adopted by many non-European countries. Other countries follow these regulations to a large extent (exception: USA).

The USA has the Federal Motor Vehicle Safety Standards (FMVSS), which constitute the technical regulations pertaining to vehicle safety and are legislative in character.

Consumer tests

The country-specific NCAP (New Car Assessment Program) is intended to provide car buyers and car manufacturers with a realistic and independent assessment of the safety features of some of the biggest-selling vehicles and to this end also conducts standardized crash tests of automobiles.

Thus, Euro NCAP is a voluntary European program for assessing vehicle safety which is supported by the European Commission and a number of European governments, and by automobile clubs and consumer organizations. Euro NCAP publishes safety reports on new cars and awards a star rating, based on the performance of the vehicles in various crash tests, including head-on, side and pole impact and collisions with pedestrians [1]. The highest rating is five stars. The requirements imposed on the star rating increase in parallel with vehicle development so that a 5-star car ten years ago is not comparable with a 5-star car today.

References

[1] www.euroncap.com/en/euro-ncap/how-to-read-the-stars/

Occupant-protection systems

In the event of an accident, occupant-protection systems are intended to keep the accelerations and forces that act on the passengers low and thereby lessen the consequences of the accident.

Seat belts with belt pretensioners provide a substantial proportion of the protective effect, absorbing the majority of the kinetic energy of the occupants. Together with a front airbag, the energy in the crash must be reduced in such a way that the occupant is not propelled up against the steering wheel or the dashboard.

In order to achieve optimum protection, the response of all components of the occupant-protection system must be adapted to one another. This is made possible by suitable sensors and high-speed signal processing. Optimum occupant protection against the effects of an impact is obtained through the precisely co-ordinate interaction of electronically fired, pyrotechnically inflated front airbags and seat-belt pretensioners. To maximize the effect of both protective devices, they are activated by the airbag ECU (triggering unit) installed in the passenger cell.

Figure 1 provides an overview of the occupant-protection systems in a passenger car.

Functions

Environment sensing

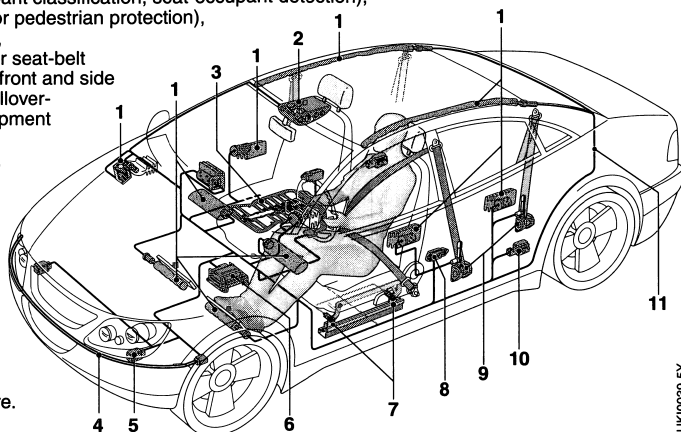
The ECU detects the different types of impact (e.g., head-on, oblique, offset, pole, rear-end, side impact) in the vehicle longitudinal direction (x -axis) and vehicle transverse direction (y -axis) with the aid of integrated acceleration sensors or structure-borne noise sensors. From their signals the crash severity and crash type to be expected is determined so that the appropriate retention systems can be activated at the right time.

Peripheral acceleration and pressure sensors installed on the vehicle body are connected via leads to the ECU. They facilitate earlier detection of vehicle contact and hence improved performance with regard to deployment of the retention systems. Especially in the case of side impacts with a small crumple zone, high-speed pressure sensors installed in the door have an advantage over acceleration sensors.

Rollover detection is performed by yaw and acceleration sensors. Aside from the yaw rate about the longitudinal axis, the vehicle lateral acceleration (y -axis) and

Figure 1: Occupant-protection systems

- 1 Airbag with gas inflator,
- 2 Passenger-compartment camera,
- 3 OC mat (occupant classification, seat-occupant detection),
- 4 Tube sensor (for pedestrian protection),
- 5 Upfront sensor,
- 6 Central ECU for seat-belt pretensioners, front and side airbags, and rollover-protection equipment with integrated rollover sensor,
- 7 iBolt,
- 8 Peripheral pressure sensor,
- 9 Seat-belt pretensioner with propellant charge,
- 10 Peripheral acceleration sensor,
- 11 Bus architecture.



the acceleration in the direction of the vertical axis (z -axis) are sensed.

The analysis algorithms in the ECU read in the data from the sensors permanently and compute whether an accident is occurring.

Activation of retention systems

The sensor signals are evaluated by intelligent algorithms, the parameters of which are optimized specifically for individual vehicles with the aid of simulations and crash tests. The airbag should not trigger from a hammer blow in the workshop, gentle impacts, bottoming out, or driving over curbstones or potholes.

The retentions systems connect to the ECU are activated when the trigger thresholds are exceeded. Current pulses activate pyrotechnic firing elements, which trigger the airbags or other retention systems.

In the event that the power supply to the ECU is interrupted by the vehicle battery as a result of the crash, a built-in power reserve provides the necessary availability.

The firing of retention systems for the relevant seat occurs only when the seat-belt buckle is connected; this is monitored by a seat-belt buckle switch.

Special functions

Advanced Rollover Sensing

Advanced Rollover Sensing uses signals from the vehicle-dynamics sensors (driving-dynamics control) to better detect soil-trip rollover processes. These data are used by the airbag ECU to calculate the speed vector and the lateral velocity. From these the deviation of the vehicle-motion vector from the vehicle's longitudinal axis and thereby lateral vehicle motion are determined.

Early Pole Crash Detection

Early Pole Crash Detection also uses the signals from the vehicle-dynamics sensors to better detect a side-on pole impact. In the event of a side-on pole impact the vehicle skids before the impact; this is detected with the yaw sensor. The information regarding the lateral movement of the vehicle is then used in the triggering algorithm to accelerate triggering of the side airbags.

Secondary Collision Mitigation

In the event of accidents the initial collision may be followed by further collisions by the accident vehicle, for example as a result of the driver losing control of the vehicle. This phenomenon puts both the vehicle occupants and other road users at risk. The Secondary Collision Mitigation function provides assistance in the event of such accidents. If an impact occurs, the airbag ECU transmits this information to the driving-dynamics control ECU. The driving-dynamics control uses specifically targeted brake interventions to slow down the vehicle or where necessary to shut down the engine. Consequential accidents can thus be avoided or their severity reduced.

Further safety measures

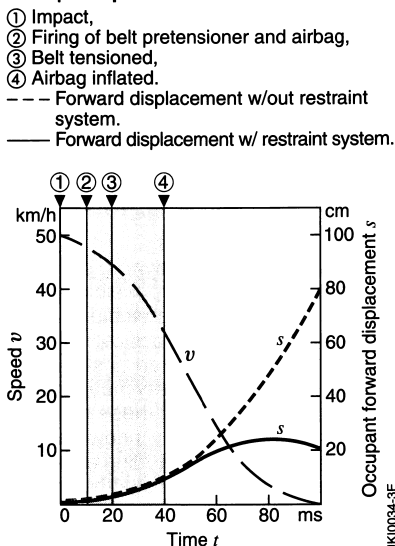
By networking driving-dynamics control and airbag ECU, it is possible also to utilize the detection of unstable or critical states to initiate more specifically targeted safety measures. When an unstable driving state is detected, safety measures can be introduced in stages; these measures can include closing the windows and the sliding sunroof, and tightening a reusable (reversible) motor-driven seat-belt pretensioner.

Retention systems and actuators

Seat belts and belt pretensioners

The function of seat belts is to restrain the occupants of a vehicle in their seats when

Figure 2: Deceleration to standstill and forward displacement of an occupant at an impact speed of 50 km/h



the vehicle impacts against an obstacle. In this way, the occupants are already involved at an early stage in the deceleration of the vehicle during an impact (Figure 2).

The standard type is the three-point belt with inertia-reel device. The belt buckle and the inertia-reel device are mounted on the seat or on the body (Figure 3). A loosely fitting belt (for example, when thick winter clothing is worn) results in belt slack. The softness of the clothing prevents the occupant from being involved at an early stage in the deceleration of the vehicle in the event of a collision. The occupant initially continues moving without being restrained, which reduces the protective effect of the belt. Furthermore, the slowed-down effect of the belt locking in an accident and the belt-webbing elongation contribute to belt slack.

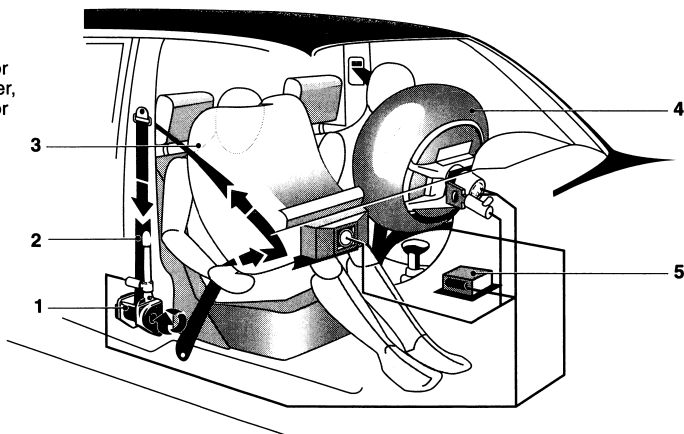
Because of belt slack three-point inertia-reel belts only afford limited protection in the event of a frontal impact at speeds in excess of 40 km/h against solid obstacles, because they cannot safely prevent the head and body from striking the steering wheel and the dashboard.

Shoulder-belt pretensioner

When activated, the shoulder-belt pretensioner eliminates the belt slack and the "film-reel effect" by rolling up and tight-

Figure 3: Occupant-protection systems with seat-belt pretensioner and front airbags

- 1 Inertia-reel device with seat-belt pretensioner,
- 2 Seat belt,
- 3 Front airbag for front passenger,
- 4 Front airbag for driver,
- 5 ECU.



ening the belt webbing. Here, the system electrically fires a pyrotechnic propellant charge (Figure 4). The gas charge released in this process acts on a plunger, which turns the belt reel via a steel cable in such a way that it is held tightly against the occupant's body. The belt webbing is therefore tightened already before the occupant starts their forward movement. With these belt pretensioner the belt webbing can be pulled back within a period of 10 ms by up to 12 cm.

Buckle tightener

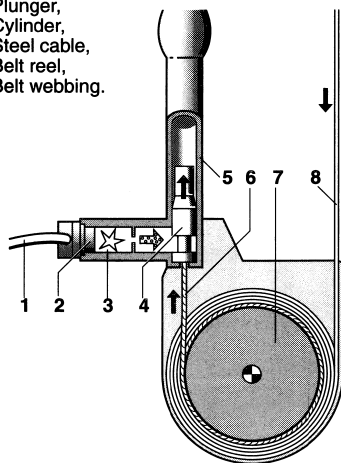
When triggered by a propellant charge or by spring systems, the buckle tightener pulls the seat-belt buckle back and simultaneously tightens the shoulder and lap belts. It improves the restraining effect and the protection to prevent occupants from sliding forward under the lap belt ("submarining effect").

Combination of shoulder-belt and buckle tighteners

The combination of both systems offers greater tightener travel to obtain an improved restraining effect. The two tighteners are activated for the most part simultaneously or with a slight time lag.

Figure 4: Shoulder-belt pretensioner

- 1 Firing wire,
- 2 Firing element,
- 3 Propellant charge,
- 4 Plunger,
- 5 Cylinder,
- 6 Steel cable,
- 7 Belt reel,
- 8 Belt webbing.



Belt-force limiter

The maximum force of the belt is limited in order to limit the forces on the occupant's torso in the event of serious crashes. Otherwise the occupant may suffer broken collarbones and ribs with resulting internal injuries.

This can on the one hand be performed by a mechanical belt-force limiter, which limits the belt force to around 4 kN. This is effected for example by the deformation of a torsion bar in the belt-retractor shaft or a deployment control seam in the belt.

An additional protective effect is provided by an electronically controlled belt-force limiter, which after reaching a defined forward displacement reduces the belt force by activating a firing element to 1 to 2 kN.

Airbags and gas inflators

Front airbags

During an accident front airbags protect the driver and the front passenger against head and chest injuries (Figure 3) which can occur when the head strikes the steering wheel or the dashboard.

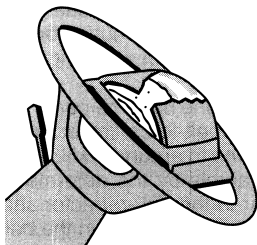
To perform this function, airbags have, depending on the installation location, type of vehicle, and structure-deformation behavior of the vehicle body, different filling capacities and shapes adapted to the vehicle conditions.

Pyrotechnic gas inflators inflate the driver and passenger airbags using dynamic pyrotechnics after a vehicle impact detected by the acceleration sensors (Figure 5). In order to achieve maximum protection, the airbag must be fully inflated before the occupant comes into contact with it. When the occupant comes into contact with the airbag, the airbag is partially deflated via deflation vents. The impact energy acting on the occupant is "softly" absorbed with noncritical (in terms of injury) surface pressures and deceleration forces.

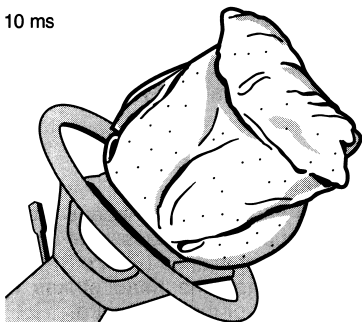
The inflation speed and the hardness of the inflated airbag can be influenced in the case of two-stage gas inflators by a delayed firing of the second stage, which injects additional charge of air into the airbag in the event of high crash severity. The maximum permitted forward motion of the driver until the airbag on the driv-

Figure 5: Highly dynamic deployment of the driver airbag

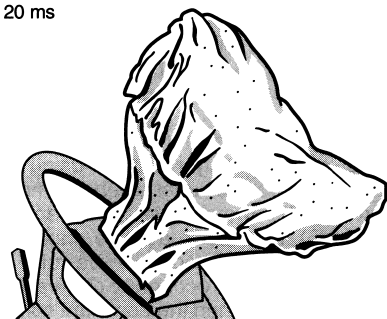
1 ms



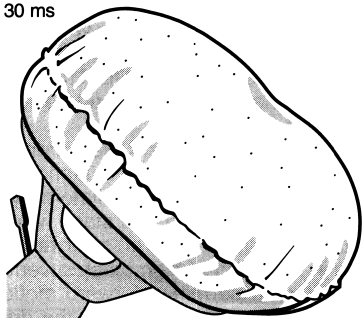
10 ms



20 ms



30 ms



UK0045-1Y

er's side has inflated is approx. 12.5 cm. This corresponds to a time of approximately 40 ms in the case of an impact at 50 km/h on a hard obstacle. 10 ms after the impact the gas inflator is activated and after 30 ms the airbag is inflated. The airbag deflates after another 80 to 100 ms through the deflation vents.

Depowered airbags

So as not to put children and very light adults on the front passenger seat at risk from the inflation process, "depowered airbags" with a reduced inflation force are used above all in the USA.

To this end, in less serious accidents at reduced speed up to around 30 km/h the airbags are inflated with 20 to 30 % less gas-inflator output, reducing the inflation speed and the hardness of the inflated airbag. This reduces the risk of injury for occupants who are "out of position" (posture diverging from the normal seating position).

With the low-risk deployment method, depending on the severity of the impact, only the first front-airbag stage is fired or else both inflator stages to bring the full gas-inflator output to bear.

Airbags with active ventilation system

These airbags have a controllable deflation valve to maintain the internal pressure of the airbag constant even if an occupant falls against it and to minimize occupant trauma. A simpler version is an airbag with "intelligent vents". These vents remain closed (so that the airbag does not deflate) until the pressure increase resulting from the impact of the occupant causes them to open and allow the airbag to deflate. As a result, the airbag's energy absorption capacity is fully maintained until the point at which its motion-damping function comes into effect.

Knee airbag

In a few vehicle types, front airbags also operate in conjunction with "inflatable knee pads", which safeguard the "ride down benefit", i.e., the speed decrease of the occupants together with the speed decrease of the passenger cell. This ensures that the upper body and head describe the rotational forward motion which is needed for the airbag to provide optimum protection. Furthermore, the knee airbag prevents contact with the dashboard support and thereby reduces the risk of injury in this area.

Side airbags

Side airbags which inflate along the length of the roof lining for head protection (e.g., inflatable tubular system, window bag, inflatable curtain) or for upper-body protection from the door or seat backrest (thorax bag) are designed in the event of a side impact to push the occupants away from the impact location and facilitate a cushioning of the impact and thereby protect them from injury.

The challenge for side airbags is the shorter time available for inflation compared with front airbags. Here there is no vehicle crumple zone and the spacing between the occupant and the vehicle body is very small. The time for crash sensing and activation of the side airbags must therefore be approximately 5 to 10 ms in the case of severe side impacts. The inflation duration for thorax bags may not exceed 10 ms.

Additional airbag equipment

A further improvement in the restraining effect is optionally provided in individual vehicles by airbags integrated in the thorax section of the seat belt (air belts, inflatable tubular torso restraints or bag-in-belt systems), which reduce the risk of broken ribs.

Inflatable headrests protect against whiplash and cervical-vertebrae injuries (adaptive headrests), inflatable carpets on the other hand protect against foot and ankle injuries. Active seats, in which an airbag in the front section of the seat surface is inflated, impede the occupant's forward sliding movement (submarining effect).

Gas inflators

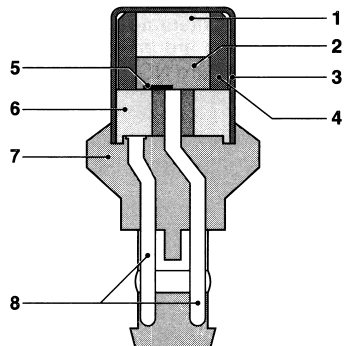
The gas inflator fills the airbag with gas and actuates the belt pretensioner. The pyrotechnic gas inflator contains a firing pellet, which when activated in turn ignites a solid propellant. The gas passes through a metal filter and the inflator's outlet ducts into the airbag.

The firing pellet (Figure 6) contains a reservoir holding the propellant charge and a priming wire. The firing pellet is connected to the airbag ECU via the connector pins and a two-wire circuit. To trigger the airbag, the ECU uses firing output stages to generate a current, which flows inside the firing pellet through the priming wire. This wire glows and activates the propellant charge.

The driver airbag fitted in the steering-wheel hub (volume approximately 60 liters) and the passenger airbag fitted in the glovebox space (approximately 120 liters) are inflated approximately 30 ms after firing.

Figure 6: Firing pellet

- 1 Propellant charge,
- 2 Detonator,
- 3 Cap,
- 4 Charge holder,
- 5 Priming wire,
- 6 Firing head,
- 7 Housing
- 8 Connector pins.



Enhanced protective functions

Rollover protection

If a car turns over, there is the danger that non-belted occupants may be thrown through the side windows, or that parts of the bodies of belted occupants (e.g., arms) may protrude from the vehicle and incur serious injury. To provide protection in such cases, existing restraint systems such as seat-belt pretensioners and side and head airbags are activated. In convertibles, the extendable rollover bars or the extendable head restraints are triggered.

A rollover is sensed in the vehicle transverse and vertical directions (y - and z -axes) with micromechanical yaw sensors and high-resolution acceleration sensors integrated in the airbag ECU. The yaw sensor is the main sensor, while the y - and z -axis acceleration sensors are used both to check plausibility and to detect the type of rollover (slope, gradient, curb-impact or soil-trip rollover).

Depending on the rollover situation, yaw rate and lateral acceleration, the retention systems and actuators are activated and deployed after 30 to 3000 ms.

Pedestrian protection

This function is intended to avoid injuries as far as possible in the event of vehicles colliding with pedestrians. Leg and head injuries can be reduced by appropriate designs of bumpers and engine hood. In addition, in a crash situation the hood can be lifted or an external airbag can cushion the impact (Figure 7). The sensor technology for this purpose (acceleration sensors or a tube with pressure sensors, see Figure 1) is integrated in the bumper. Appropriate tests are an integral part for example of the Euro NCAP crash test.

Occupant sensing

There are various options on the market for occupant classification.

iBolt

Occupant classification can be performed with the iBolt ("intelligent" bolt), an absolute-weight-measuring method. These iBolts (see Figure 1) measure the forces involved and secure the seat frame (suspended seat) on the sliding base, replacing the four mounting screws that are otherwise used. They measure the weight-dependent change in the gap between the bolt sleeve and the internal bolt with integral Hall-element IC connected to the sliding base (Figure 8).

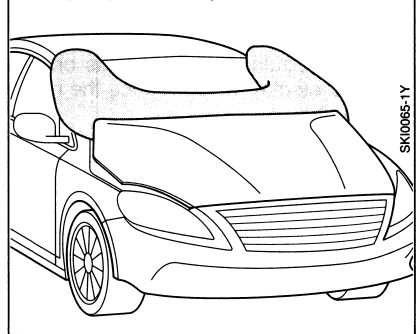
Seat-occupancy mats

Seat-occupancy mats combined with pressure-sensitive elements and matching sensor also perform this function (Figure 9). The information is sent to the ECU and incorporated in the triggering strategy for the means of protection. The seats can be adjusted when backrest angle and position are sensed with suitable actuators during the crash to better protect the occupants.

Deactivation of front-passenger airbag

A manually actuated deactivation switch can be used to disable the passenger airbag. This is required if certain child seats are used in this space.

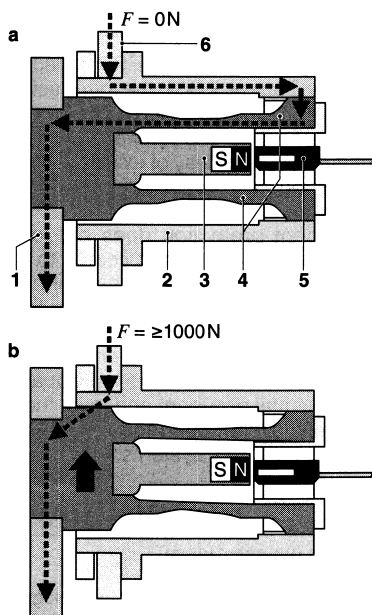
Figure 7: Raised engine hood and external airbag as pedestrian protection



SK10065-1Y

Figure 8: Force-measuring iBolt

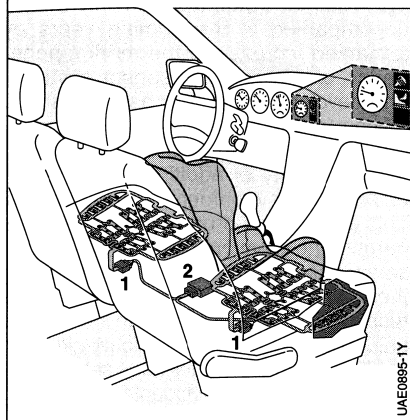
- a) Rest position,
 b) In operation, i.e., in overload stop.
 1 Sliding base,
 2 Sleeve,
 3 Magnet holder,
 4 Double bending beam (spring),
 5 Hall IC,
 6 Seat frame.



There is an increasing availability of standardized anchoring systems (ISOFIX child seats). Switches integrated in the anchoring locks initiate an automatic front-passenger airbag deactivation, which must be indicated in the instrument cluster.

Figure 9: Installation position of the OC sensor mats in the vehicle front seats

- 1 OC ECU,
 2 Airbag ECU.



Out-of-position detection

In order to prevent injuries being caused by airbags to occupants who are "out of position" (e.g., are leaning far forwards) or to small children in reboard child seats (rearward-facing seating position), the triggering of the front airbag and its inflation must be adapted to the situation.

Enhanced detection of the occupant position and adaptation of the retention systems and their activation are important for future automated driving, since the challenges faced by passive safety increase with the freedom of movement gained by the occupants.

Integral-safety systems

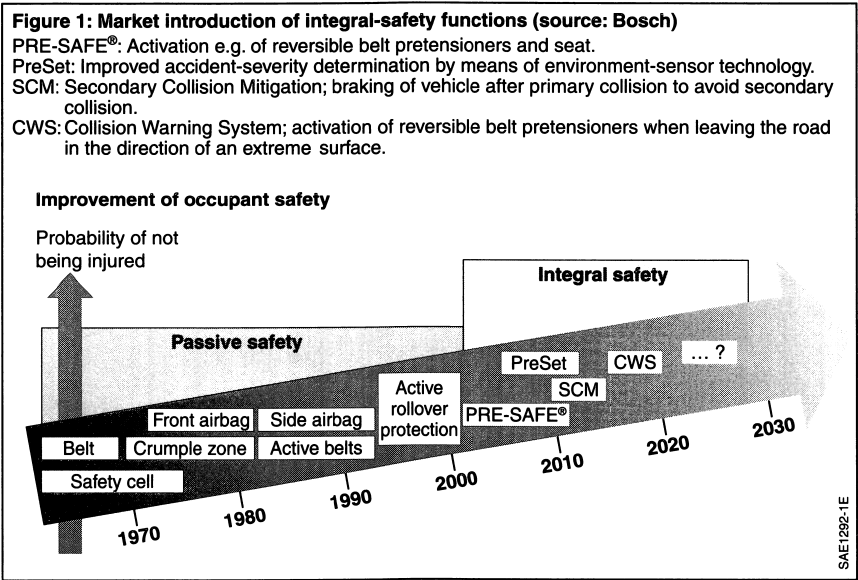
Overview

The road to automated driving will be accompanied in the coming years by a marked increase in the market penetration of environment-sensor systems. The development of new assistance and driving functions necessitated by the introduction of automated driving leads to the availability of additional information on vehicle, environment, and occupants. This extended and networked data basis permits the development of new integral safety concepts with the aim of adapting the vehicle's protective measures to the respective accident situation in the event of unavoidable accidents. Predictive situation analysis in particular facilitates safety measures for preventing accidents or lessening the accident severity. The networking of active and passive safety measures creates the precondition for situational adaptation of the vehicle's automatic protective responses to the respective accident events.

Motivation

It has been possible in the past few years through numerous measures in the automotive industry, traffic research and legislation to reduce the number of traffic fatalities and serious injuries. The fitting prescribed by law of all new passenger cars with driving-dynamics control (Electronic Stability Program, ESP®; Electronic Stability Control, ESC), the introduction of antilock braking systems (ABS) for motorcycles, and lane-change warning systems are mentioned by way of example. Numerous serious accidents still occur despite these measures [1].

Accident protection does not lose in importance even when it comes to automated driving. There is a need for action, since integral safety forms the fallback level of automated driving. The integral-safety approach offers new solutions which will both take into account the individuality of the individual occupant and raise the traffic-safety level through the networking of sensor and safety systems.



Technology of the present day

Integral safety is based on the networking of components and algorithms of various safety and comfort and convenience systems in the vehicle (e.g., passive-safety retention systems, driving-dynamics control, driver-assistance systems, passenger-compartment sensing). Here, the entire chain of events of an unavoidable accident before, during and after the collision is analyzed to minimize the risk of injury to vehicle occupants and other road users.

The first integral-safety systems were launched in 2002 by Mercedes-Benz under the name PRE-SAFE® in the S-Class. Since 2009 the front close-range radar sensors of the driver-assistance package have also been used with this system to detect potential collision objects with preventive reversible seat-belt tightening. In 2013 this principle was carried over to the vehicle rear end.

A further step in the direction of integral safety was made by the "PRE-SAFE impulse side" function introduced in 2016 for the front seats, in which for the first time pyrotechnics are activated exclusively on the basis of environment-sensor technology. Side-mounted radar-sensor technology detects an unavoidable side impact, whereupon an actuator integrated in the seat imparts to the occupant an impulse towards the center of the vehicle [2].

Figure 1 shows the timeline of the transition from classic passive safety to integral safety. Safety functions and safety components already introduced in the market are depicted.

Solution approaches

Challenges

Since the introduction of radar technology for driver-assistance functions like Adaptive Cruise Control (ACC) or Automatic Emergency Braking (AEB) in the automobile sector, new function ideas have arisen which are founded on the use of environment data to better estimate the accident severity (e.g., the PreSet function for determining accident severity). However, such functions have not been widely used due to the low equipment rate with Adaptive Cruise Control. This situation is changing, as Automatic Emergency Braking has since 2014 been rewarded in the Euro NCAP protocol. As a result, the proportion (equipment rate) of new cars fitted with radar sensors has increased. In spite of this conducive boundary condition there are hurdles for the market introduction of new integral-safety systems:

- Effects of new safety functions only become visible in the accident statistics after a delay, as the rising equipment rate in new-vehicle registrations is set against an old vehicle population.
- Dedicated environment sensors and processing units only for the integral-safety functions have not gained acceptance in the past for cost reasons, which hinders the introduction of independent integral-safety systems.
- Networked functions with numerous data interfaces between assistance and restraint systems entail additional application and validation outlay.

The penetration times of integral-safety systems will follow the market developments with regard to driver assistance, active safety, and automated driving. In the course of the introduction of automatic driving functions the equipment rate with environment and passenger-compartment sensors and with domain ECUs will increase [3]. The introduction of automatic driving functions will ensure the market introduction and market penetration of environment and passenger-compartment sensors and of appropriate domain ECUs. At the same time, occupant protection faces hitherto unknown challenges as a result of these functions. Possible new seating positions,

shifting of the occupant positions during automatic braking or steering operations or larger objects between occupants and airbags change the boundary conditions for the optimum use of retention systems in the event of a collision risk.

Even fundamental elements of present-day vehicles, such as function and E/E architecture, undergo a development process in the course of developments in the fields of electromobility, automated driving, and connectivity. New requirements with regard to safety, communication speed, computing outlay, latencies, and costs drive this development process, which is also useful in the introduction of integral-safety systems.

Application examples

Accident types with head-on collision

Head-on-accident scenarios as depicted in Figure 2 highlight an important sphere of activity for integral safety. The frequently occurring rear-end collision (Figure 2a) can already today be firmly detected in terms of its development by the use of environment-sensor technology. The growing number of vehicles which are equipped with an automatic emergency brake system will reduce the frequency and severity of this type of accident.

Intersection accidents with side collisions as pictured in Figure 2e constitute a further area of use for integral-safety functions. Injury severity is frequently in the middle range. Intersection-assistance systems with left turn-off assist and cross-traffic assist will greatly reduce the frequency of such accidents (see Driver-assistance systems).

The introduction of Electronic Stability Programs, lane-departure warning systems, and functions for detecting driver fatigue brought about a reduction of in collisions caused by the vehicle leaving the road (e.g., tree collision, Figure 2d). This accident type is nevertheless still very relevant today on account of the injury severity often associated with it.

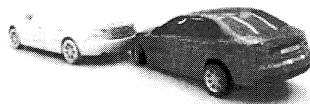
Accidents involving oncoming vehicles exhibit a high injury probability (Figure 2c). Active-safety and automation measures prove to be the least effective in these accident scenarios. Integral safety is therefore focused on this accident type.

There is need for improvement in the case of accident types which are not reproduced by laboratory tests or for which conventional contact-sensor technology exhibits limitations. These conditions apply for example to driving under a truck

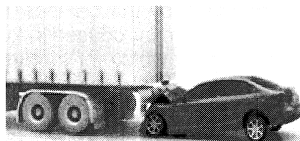
Figure 2: Relevant application cases in the area of head-on collisions (source: Bosch)

- a) Rear-end collision,
- b) Driving under a truck,
- c) Head-on collision,
- d) Tree collision,
- e) Side collision.

a



b



c



d



e



(Figure 2b) or a central impact with a tree with a small overlap (Figure 2d). In both cases the sensors installed in the supporting frame respond only at a relatively late stage after the body has undergone marked deformation.

Function examples

The risk of injury increases if the moment of airbag deployment is not optimally matched to the forward displacement of the occupant. This situation ensues when the airbag is fired too late or when the occupant is too close to the airbag due to forward displacement caused by a braking operation ([4], [5]). Reference is made by way of example to two functions which address this problem: Integrated Collision Detection Front (IDF) and Pre-Crash Positioning (PCP).

Integrated Collision Detection Front (IDF)

The IDF function uses environment-sensor data (radar, video) in conjunction with conventional contact-sensor technology to trigger belt pretensioners and airbags at an earlier stage. Aside from an accurate estimation of the relative impact speed and the moment of impact, the merged sensor system offers the following advantages in terms of optimizing the efficiency of the retention systems:

- more precise description of the collision object,
- optimization of the interfaces between ADAS (Advanced Driver Assistance Systems) and restraint system.

Pre-Crash Positioning (PCP)

The PCP function coordinates in scenarios depicted in Figure 2 automated emergency braking with the restraint system. The chronological sequences of forces acting on the occupants, braking operations, and activations of the retention systems must be matched in such a way as to minimize the load on the occupants. A first instance of this function serves to position the occupants at the right time in the phase prior to the collision by means of reversible seat-belt tightening.

Further developments

Later versions of the IDF and PCP functions will, along similar lines to the further development of the automatic emergency

brake systems, take into account more complex relationships. The first expansion stage handles the rear-end impact using front-sensor technology with the merging of radar and video. The same sensor configuration is used in the case of oncoming vehicles. However, this accident type requires the development of more complex decision algorithms and additional safeguarding effort to avoid bad decisions. Expansion to include intersection accidents necessitates the additional installation of Mid-Range-Radar (MRR) with a very large side beam width.

Individual safety

On account of legal regulations and consumer tests today's passive-safety systems are optimized for certain occupant heights and weights. Adaptive retention systems combined with suitable activation offer the potential for raising the safety level also for occupants that deviate from the standard sizes used in the tests. One possible solution approach uses occupant-specific variables such as age, height, weight or gender for the purpose of individually activating existing and future adaptive retention systems.

These data are acquired via passenger-compartment-sensor technology, supplemented by user inputs and plausibility checks. For example, the user enters once via his smart phone or an HMI (Human Machine Interface) integrated in the vehicle his personal data, which are linked with an image recorded by him. On each new entry into the vehicle a unique identification of the occupant is performed by means of passenger-compartment camera and image-processing algorithms with corresponding parameterization of the retention systems. If the occupant is not identified, the usual high safety level is still available in the default settings.

In view of the anticipated increase in the use of automatic and semi-automatic driving functions, an increase in the number of passenger-compartment cameras being installed is also to be expected. Personal optimization of the safety functions therefore does not require additional sensors. It is possible to fall back on passenger-compartment-sensor technology to be introduced to safeguard automated driving.

Safeguarding

A major challenge faced by integral-safety functions is environment and passenger-compartment sensing and correct situation assessment. As well as verifying correct functionality, particular attention is placed on safeguarding against unwanted false triggering of the retention systems. A typical situation that is to be avoided is the erroneous reporting of a collision object without such an object actually existing.

A further scenario to be avoided is as follows: An object is correctly detected, however an erroneous trajectory computation suggests a collision course and activates retention systems although the object is actually driving past the system vehicle. Falling into the same category are driving situations in which an object on a collision course is correctly detected, but after retention systems have been triggered the collision is not avoided [2].

In spite of data fusion and the associated redundancy, there are complex situations whose detection and correct interpretation

must be verified. To guarantee this, the testing of integral-safety systems must record as wide a variety of real traffic situations as possible.

Testing of integral-safety system according to today's methodology exceeds both the cost and the time framework for release in the automotive industry. A first step towards reducing expenditure while simultaneously guaranteeing safety is to break down the wide range of real situations into a limited number of simulated test-track or laboratory situations which are representative of reality.

A supplementing approach is to use virtual instead of real test environments. Virtual test environments permit the evaluation of safety functions in the early development phase and thus fast prototype construction. Furthermore, they enable safeguarding of the functions to be developed in complex traffic situations that are hard to reproduce in real driving or test operation.

Outlook

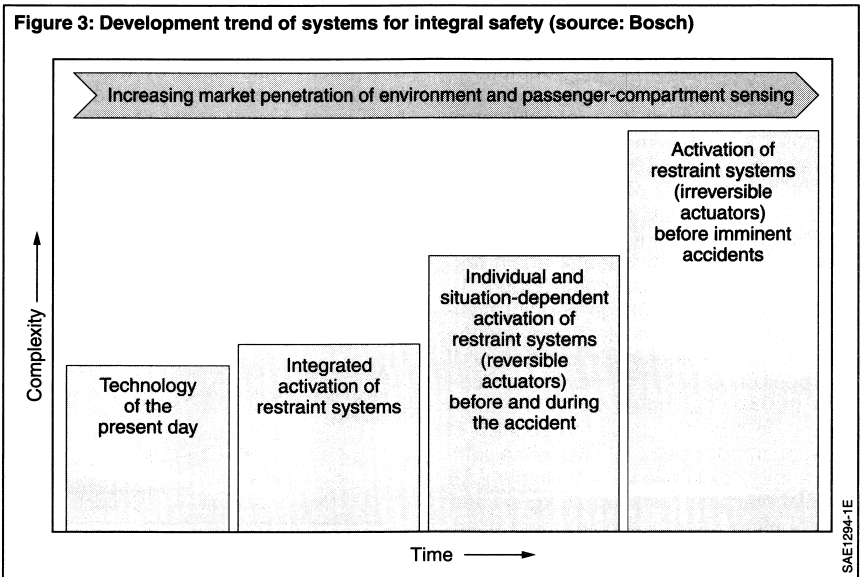
Starting out from conventional vehicle safety with deployment of the airbags after an impact has been detected, the trend is towards adaptation of the trigger thresholds of retention systems on the basis of networked object data by environment sensors, passenger-compartment sensors, and driving-dynamics information. In this expansion stage of the integral-safety systems only reversible protection systems will be activated prior to the collision.

Functions with activation of irreversible actuators exclusively on the basis of data from environment sensors will constitute the next development stage of integral safety (Figure 3).

References

- [1] T. Lich et al.: "Is there a broken trend in traffic safety in Germany? Model based approach describing the relation between traffic fatalities in Germany and environmental conditions", ESAR Conference, Hanover, Germany, 2014.
- [2] J. Richert, R. Bogenrieder, U. Merz, R. Schöneburg: "PRE-SAFE® Impuls Seite – Vorauslösendes Rückhaltsystem bei dro-
- hendem Seitenaufrall – Chance für den Insassenschutz, Herausforderung der Umfeldsensorik", 10th VDI Conference Vehicle Safety Safety 2.0, Berlin, 2015.
- [3] J. Becker, S. Kammel, O. Pink, M. Fausten: "Bosch's approach toward automated driving". at–Automatisierungstechnik 2015; 63(3): 180–190.
- [4] G. Gstrein, W. Sinz, W. Eberle, J. Richert, W. Bullinger: "Improvement of airbag performance through pre-triggering". Paper Number 09–0229, Proceedings of 18th Enhanced Safety of Vehicles, ESV 21, Stuttgart, Germany, 2009.
- [5] H. Freienstein, T. Engelberg, H. Bothe, R. Watts: "3-D Video Sensor for Dynamic Out-of-Position Sensing, Occupant Classification and Additional Sensor Functions". Paper No. 2005-01-1232, SAE World Congress 2005, Detroit, USA.

Figure 3: Development trend of systems for integral safety (source: Bosch)



Vehicle bodies, passenger cars

Main dimensions

Interior dimensions

The dimensional layout depends on body shape, type of drive, scope of aggregate equipment, desired interior size, trunk volume, and other considerations such as driving comfort, driving safety, and operating safety (Figure 2). The seating positions are designed according to ergonomic findings and with the aid of templates or 3D CAD dummy models (DIN, SAE, RAMSIS): body template as per DIN 33408 [1], for men (5th, 50th, and 95th percentile) and women (1st, 5th and 95th percentile). For example, the 5th percentile template represents "small" body size, i.e. only 5% of the population has smaller bodies, while 95% has larger body dimensions.

SAE H-point template in accordance with SAE J826 [2] (May 1987): 10th, 50th, and 95th percentile thigh segments and lower leg segments. For legal reasons, motor-vehicle manufacturers in the ECE, the USA and Canada must use the SAE H-point template for determining the seating reference point. Most motor-vehicle manufacturers around the world use the 3D CAD dummy model RAMSIS (computer-aided anthropological-mathematical system for passenger simulation) in development.

The hip point (H-point) is the pivot center of torso and thigh, and roughly corresponds to the hip-joint location. The seating reference point (SRP, ISO 6549 [3] and US legislation) or R point (ISO 6549 and EEC Directives/ECE regulations) is the position of the H-point in the seat adjustment field on variable seats, again taking account of the heel point (Figure 1). In determining the design H-point position, many vehicle manufacturers use the 95th percentile adult-male position or, if this position is not reached, the position with the seat adjusted to its rearmost setting. In order to check the position of the measured H-point relative to the vehicle, a three-dimensional, adjustable SAE H-point machine weighing 75 kg is used. The seating reference point, heel point, vertical, and horizontal distance between these two points, and body angles speci-

fied by the vehicle manufacturer form the basis for determining the dimensions of the driver's seating position.

The seating reference point is used:

- To define the positions of the eye ellipse (SAE J 941, [4]) and the eye points (RREG 77/649, [5]) as a basis for determining the driver's direct field of view.
- To define hand reach envelopes in order to correctly position controls and actuators.
- To determine the accelerator heel point (AHP) as a reference point for positioning the pedals.

The space required by the rear axle as well as location and shape of the fuel tank primarily determine the rear-seating arrangement (height of the seating reference point, rear seating room, headroom) and thus the shape of the roof rear portion. Depending on the type of vehicle under development, the projected main dimensions and the required passenger sizes, there are different body angles for the 2D templates or body postures (RAMSIS) and different distances between the seating reference points of the driver's and rear seats. The longitudinal dimensions are greatly influenced by the height of the seating reference point above the heel point. Lower seats require a more

Figure 1: Key parameters of car-cockpit design

Normal hand position: bottom edge of hand level with steering wheel.

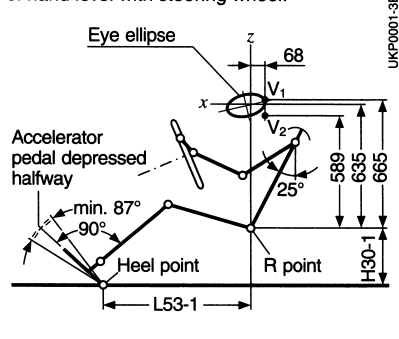
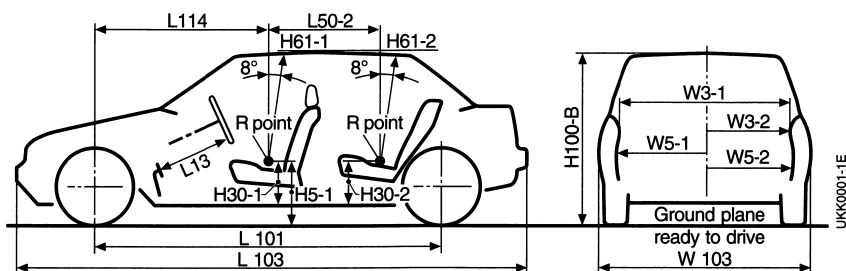


Figure 2: Typical internal and external dimensions (as per DIN 70 020, Part 1)

Dimension		Subcompact	Luxury class
		Distance in mm	mm
H5-1	R point to datum plane at front	460	680
H30-1	R point to heel point at front	240	310
H30-2	R point to heel point at rear	300	315
H61-1	Effective head room at front	940	1,010
H61-2	Effective head room at rear	900	1,010
H 100-B	Vehicle height	1,380	1,660
L 13	Steering wheel to brake pedal	500	630
L50-2	R point distance (front to rear seat)	710	830
L 101	Wheelbase	2,500	3,000
L 103	Vehicle length at all points	3,840	4,930
L 114	Center of front wheel to R point	1,250	1,600
W3-1	Shoulder room at front	1,300	1,540
W3-2	Shoulder room at rear	1,290	1,520
W5-1	Hip room at front	1,300	1,500
W5-2	Hip room at rear	1,300	1,500
W103	Vehicle width at all points	1,630	1,930

stretched passenger seating position, and thus greater interior length.

The passenger-cell width, and thus the shoulder room, elbow room and hip room, are dependent on the projected external width, the shape of the sides (curvature), and the space required for door mechanisms, passive restraint systems, and various assemblies (propshaft tunnel, exhaust-gas system, etc.).

Trunk dimensions

The size and shape of the trunk are dependent on the design of the rear end of the vehicle, the position of the fuel tank and its volume, the entire rear package, the positions of the axle and the resulting wheel arches, and the location of the main muffler.

The trunk capacity is determined as per DIN ISO 3832 [6] or, more commonly, us-

ing the VDA (Association of German Engineers) method with the VDA module (cuboid $200 \times 100 \times 50 \text{ mm}^3$ – corresponds to 1 dm^3 volume).

External dimensions

The following factors must be taken into consideration:

- seating arrangement and trunk,
- engine, transmission, radiator,
- auxiliary systems and special equipment,
- space requirement for compressed or pivoted wheels (allowance for snow chains),
- type and size of powered axle,
- position and volume of fuel tank,
- front and rear fenders,
- aerodynamic considerations,
- ground clearance (approx. 100 to 180 mm),
- visibility legislation.

Body design

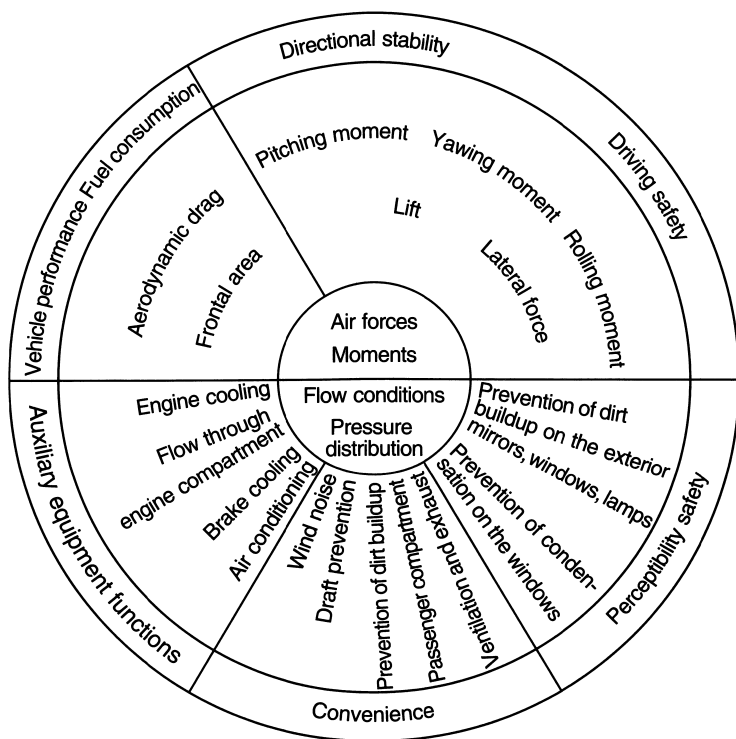
The following technical requirements must be met in interior and exterior body design:

- Mechanical functions (lowering of side windows, opening of hood, trunk lid, and sliding sunroof, positions of lamps).
- Manufacturability and ease of repair (gap widths, bodywork assembly, window shape, protective molding rails, paint feature lines).
- Safety (position and shape of fenders, no sharp edges or points).
- Aerodynamics (air forces and moments if they affect performance, fuel consumption, and emissions as well as

vehicle dynamics/directional stability, wind noise, adhesion of dirt to the outer body panels, enjoyment of open-topped driving, cabin ventilation, windshield wiper function, cooling of fluids and components; see vehicle aerodynamics, also see Figure 3).

- Optics (visual distortion caused by window type and slope, glare due to reflection).
- Legal requirements (position and size of lamps, rearview mirror, license plates).
- Design and layout of controls (positions, shapes, and surface contours).
- Visibility of vehicle extremities (parking).

Figure 3: Aerodynamic effects on vehicle operations and vehicle characteristics



Body

Body structure

Unitized body

The unitized body (standard design) consists of sheet-metal panels, hollow tubular members, and body panels that are joined together by multi-spot welding machines or welding robots. Individual components may also be bonded, riveted, or laser-welded.

Depending on vehicle type, roughly 5,000 spot welds must be made along a total flange length of 120 to 200 m. The flange widths are 10 to 18 mm. Other parts (front fenders, doors, hood, and trunk lid) are bolted to the body supporting structure. Other types of body construction include frame and sandwich designs.

Increasing use is being made of hybrid bodyshell construction, where the individual structural components of the bodyshell are made of different materials according to function and required load capacity. The bodyshell illustrated in Figure 4 (MBW 211), for example, consists of 47% high-strength steel, 42% standard steel, 10% aluminum, and 1% plastic.

The general requirements placed on the body structure are as follows:

Rigidity

Torsional and bending rigidity should be as high as possible in order to minimize elastic deformation of the apertures for the doors, hood, and trunk lid. The effect of body rigidity on the vibrational characteristics of the vehicle must be taken into consideration.

Vibrational characteristics

Body vibrations as well as vibrations of individual structural components as a result of excitation by the wheels, wheel suspension, engine, and drivetrain can severely impair driving comfort if resonance occurs.

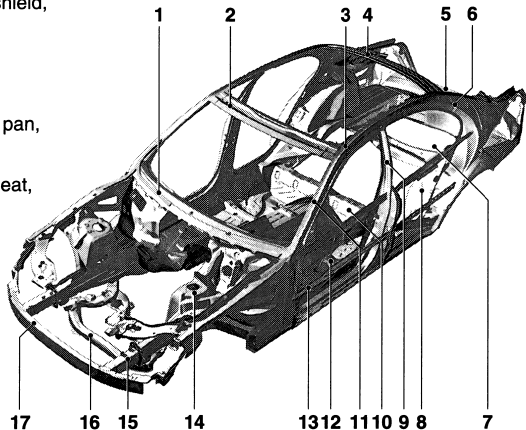
The natural frequency of the body, and of those body components which are capable of vibration, must be detuned by means of creasing (impressions in the component to stiffen flat structures), and changing the wall thicknesses and cross sections to minimize resonance and its consequences.

Operational integrity

The alternating stresses that occur during vehicle operation place a strain on the body over time. The layout of the body (focal points are mounting points of the

Figure 4: Body structure

- 1 Cross member below windshield,
- 2 Roof frame, front,
- 3 Roof frame, side,
- 4 Roof frame, rear,
- 5 Rear-facing panel,
- 6 C-pillar,
- 7 Rear floor and spare-wheel pan,
- 8 Side member, rear,
- 9 B-pillar,
- 10 Cross member under rear seat,
- 11 A-pillar,
- 12 Cross member under driver's seat,
- 13 Side member,
- 14 Shock-absorber mounting,
- 15 Side member, front,
- 16 Integral cross member,
- 17 Cross member, front.



chassis, steering and drive units) must take account of the alternating stresses so that the body also remains undamaged during rough vehicle operation.

Body stresses due to accidents

In the event of a collision, the body must be capable of transforming as much kinetic energy as possible in the deformation process, while minimizing deformation of the passenger cell.

Ease of repair

The components which are most susceptible to damage as a result of minor accidents ("fender-benders") must be easily replaceable or repairable (access to exterior body panels from the inside, access to bolts, favorable location of joints, feature lines for repainting individual components).

Body materials

Sheet steel

Steel plates with different qualities are normally used for the bodyshell. The bodyshell plate thicknesses are 0.6 to 3.0 mm; the highest proportion are steel sheets of 0.75 to 1.0 mm in thickness. Due to the mechanical properties of steel with regard to rigidity, strength, economy, and ductility, alternative materials for the vehicle body structure are not yet fully available.

High Strength Low Alloy (HSLA) sheet steel is used for high-stress structural components. The resulting high strength of these components allows their thickness to be reduced.

Aluminum

In order to reduce weight, aluminum can be used for separate body components, such as the hood, trunk lid, etc.

Since 1994 an aluminum body has been in use on one of the German luxury

Table 1: Examples of alternative materials

Typical applications	Material	Abbreviation	Processing method
Supporting parts, e.g. fender spring support	Glass-mat reinforced thermoplastic	PP-GMT	Injection molding
Trim panels and finishers, e.g. front apron, spoiler, front section, radiator grill, wheel-housing cladding, hubcaps. Bodyshell parts, e.g. hood, mudguards, trunk lid, sliding sunroof	Glass-mat reinforced thermoplastic	PP-GMT	
	Polyurethane	PUR	RIM (Reaction-Injection Molding) RRIM (Reinforced- Reaction-Injection Molding)
	Polyamide Polypropylene Polyethylene Acrylonitrile-butadiene-styrene (ABS) copolymer Polycarbonate (with polybuteneterephthalate)	PA PP PE ABS PC-PBT	Injection molding, glass-fiber proportion determines elasticity
Elastic fender and front guard strips	Polyvinylchloride	PVC	Injection molding and extruding
	Ethylene-propylene terpolymers Elastomer-modified Polypropylene	EPDM PP-EPDM	
Energy-absorbing foam	Polyurethane Polypropylene	PUR PP	Reaction foams
Fenders	Duromers with reinforcing fibers (Sheet Molding Compound)	SMC	Pressing

classes. The vehicle's frame is constructed from aluminum extruded sections, and the panel parts are integrated as self-supporting parts (ASF Audi Space Frame). The implementation of this principle required the use of suitable aluminum alloys, as well as new production processes and special repair facilities. According to the manufacturers, the rigidity and deformation characteristics are identical to those of steel, or are even superior.

Plastics

Plastics replace steel in a limited number of cases for separate body components to reduce weight (see Table 1).

Body surface

Corrosion protection

Allowance must be made for corrosion protection as early as the body-design phase. The "Anti Corrosion Code", Canada, is an agreement which was developed by Scandinavian car manufacturers and consumer-protection organizations and establishes that all vehicles after January 1983 may not exhibit either surface rust for a period of three years or rust penetration or solid structural weakening for a period of six years.

Corrosion-protection measures:

- Minimize the number of flanged joints, sharp edges and corners,
- Avoid areas where dirt and humidity can accumulate,
- Provide holes for pretreatment and electrophoretic enameling,
- Provide good accessibility for the application of corrosion inhibitor,
- Allow for ventilation of hollow spaces/cavities,
- Prevent the penetration of dirt and water to the greatest extent possible; provide water drain openings,
- Minimize the area of the body exposed to stone-chip impact,
- Prevent contact corrosion.

Precoated sheet steel (inorganic zinc, electrolytically galvanized, hot-dip galvanized) is often used for those components at particularly high risk, such as doors and load-bearing members at the front of the vehicle. Inaccessible structural areas are

coated with spot-welding paste (PVC or epoxy adhesive, approx. 10 to 15 m seam length per vehicle) prior to assembly.

Painting

Measures subsequent to electrophoretic enameling:

- Covering the spot-welded seams (up to 90 to 110 m), welts and joints with PVC sealing compound
- Coating the underbody against stone-chip impact damage with PVC underbody protection (0.3 to 1.4 mm thick, 10 to 18 kg per vehicle); alternatively, using paneling sections made of plastic
- Filling cavities with penetrating, non-aging water-based wax
- Using corrosion-resistant, attached plastic components in high-risk areas, such as the front fenders (PVC coating not used in those places)
- Sealing of underbody and engine compartment after final assembly.

Body finishing components

Fenders

The front and rear of the vehicle should be protected so that low-speed collisions will only damage the vehicle slightly, or not at all. Prescribed fender evaluation tests (US Part 581 [7], Canada CMVSS 215 [8], and ECE-R42 [9]) specify minimum requirements in terms of energy absorption and installed fender height. Compliance with fender tests in the USA to US Part 581 (4 km/h barrier impact, 4 km/h pendulum tests) and in Canada (8 km/h) requires a fender system with energy absorbers that automatically regenerate. The requirements of the ECE standard are satisfied by plastic-deformable retaining elements located between the bendable bar (fender) and the vehicle body structure. In addition to sheet steel, many

Table 2: Paint/coating thicknesses

Paint application, total thickness	≈ 120 μm
Zinc-phosphate coat	≈ 2 μm
Electrophoretic enameling (cathodic)	13 to 18 μm
Filler	≈ 40 μm
Finishing paint	35 to 45 μm
Clear coat finish (only for metallic and water-based paint)	40 to 45 μm

bendable bars are made using fiber-reinforced plastics and aluminum sections.

Exterior trim panels, protective molding rails

Plastics have become the preferred materials for external protective molding rails, trim panels, skirts, spoilers, and particularly for components whose purpose is to improve the aerodynamic characteristics of the vehicle. Criteria used in the selection of the proper material are flexibility, high-temperature shape retention, coefficient of linear expansion, notched-bar toughness, scratch resistance, chemical resistance, surface quality, and paintability.

Glazing

The windshield and rear window are retained in rubber strips, and sealed or bonded in place. The weight of glazing per vehicle is 25 to 35 kg. The substitution of glass with plastics (PC, PMMA) to reduce weight has not succeeded due to various disadvantages. Laminated safety glass is sometimes used in door windows to provide better sound and heat insulation. Sliding sunroofs are also increasingly made of glass (usually single-pane toughened safety glass).

Door locks

Door locks are of immense importance to passive accident safety. Various manufacturers have produced a variety of solutions to the issues of ease of operation, theft deterrence, and child safety. The legal requirements are as follows:

ECE (ECE-R11, [10]):

Every lock must have a latched position and a fully closed position.

- Longitudinal force: capable of withstanding 4,440 N in latched position, 11,110 N in fully closed position.
- Lateral force: capable of withstanding 4,440 N in latched position, 8,890 N in fully closed position.
- Inertial force: The lock should not be forced out of the fully closed position by linear or lateral acceleration of 30 g acting on the lock in both directions, and on the striker and the operating device, when the locking mechanism is not engaged.

USA (FMVSS 206, [11]):

Every lock must have a fully closed position. Doors which are attached by hinges must have a latched position.

- Longitudinal force: capable of withstanding 4,450 N in latched position, 11,000 N in fully closed position.
- Lateral force: capable of withstanding 4,450 N in latched position, 8,900 N in fully closed position.
- Inertial force: The lock should not be forced open from the fully closed position by linear or lateral acceleration of 30 g acting on the door lock system in both directions (lock and operating device).

Trunk locks

(Excerpt from FMVSS 401 [12], in force since 9.1.2002)

These safety regulations for passenger cars with trunks contain the requirements for a release mechanism for the trunk. Such a mechanism should allow a person locked inside the trunk of a car to free himself/herself from the trunk. According to these regulations, a manually operated release mechanism must be provided with a function (e.g. lighting or phosphorescence) by which the release mechanism can easily be seen inside the closed trunk.

Seats

The strength requirements which must be met by seats in a collision pertain to the seat frame (seat cushions, backrest), the head restraints, the seat adjustment mechanism, and the seat anchors (regulations: FMVSS 207 [13], 202 [14]; ECE-R17 [15], 25 [16]; RREG 74/408 [17], 78/932 [18], and others).

One component of active safety is seating comfort. Seats must be designed so that vehicle occupants with different body dimensions are able to sit for long periods of time without becoming tired. Parameters:

- Support of individual body areas (pressure distribution),
- Lateral support when cornering,
- Seat climatic conditions,
- Freedom of movement so that an occupant can change his/her sitting position without having to readjust the seat,
- Vibrational and damping characteristics (matching the natural frequency within the excitation frequency band),
- Adjustability of seat cushion, backrest and head restraint.

The above parameters are affected by the following:

- Dimensions and shapes of the upholstery of seat cushions and backrests
- Distribution of the spring rates of individual padded zones
- Overall spring rate and damping capacity of the seat cushions in particular
- Thermal conductivity and moisture-absorption capacity of the covering material and upholstery
- Operation and range of the seat adjustment mechanisms.

Interior trim

A section of trim consists of a dimensionally stable core (sheet steel, sheet aluminum, or plastic) with fastening elements, and energy-absorbing padding made of foam material (e.g. PUR), and a flexible surface layer. One-piece plastic trim sections made of injection-molded thermoplastic material are also used.

The headliner is made either as a stretched liner or finished liner. The materials used must be flame-retardant and slow burning (FMVSS 302 [19]).

Safety

Active safety

The purpose of active safety is to prevent accidents. Driving safety is the result of a harmonious chassis and suspension design with regard to wheel suspension, springing, steering, and braking, and is reflected in optimum dynamic vehicle behavior.

Conditional safety results from keeping the physiological stress to which vehicle occupants are subjected by vibration, noise, and climatic conditions down to as low a level as possible. It is a significant factor in reducing the probability of misactions in traffic.

Vibrations within a frequency range of 1 to 25 Hz (stuttering, shaking, etc.) induced by wheels and drivetrain assemblies reach the occupants of the vehicle via the car body, seats, and steering wheel. The effect of these vibrations is more or less pronounced, depending on their direction, amplitude, and duration.

Noises produced as acoustical disturbances in and around the vehicle can come from internal sources (engine, transmission, propshafts, axles) or external sources (tires on roads, wind noises), and are transmitted through the air or by structures. The sound pressure level is measured in dB (A).

Noise reduction measures are concerned, on the one hand, with the development of quiet-running components and the insulation of noise sources (e.g. engine encapsulation), and on the other hand, with noise damping by means of insulation or anti-noise materials.

The major climatic influences are air temperature, air humidity, air flow rate, and air pressure.

Perceptibility safety

Measures which increase perceptibility safety are concentrated on:

- Lighting equipment,
- Acoustic warning equipment,
- Direct and indirect visibility (driver's view: the angle of binocular obscuration – i.e. for both of the driver's eyes – caused by the A-pillars must not be more than 6 degrees).

Operational safety

Low driver stress, and thus a high degree of driving safety, requires optimum design of the driver's surroundings with regard to ease of operation of the vehicle controls.

Passive safety

The purpose of passive safety is to mitigate the consequence of accidents.

Exterior safety

The term "exterior safety" covers all vehicle-related measures which are designed to minimize the severity of injury to pedestrians, and bicycle and motorcycle riders struck by the vehicle in an accident. The determining factors are the vehicle-body deformation behavior and the exterior vehicle-body shape.

The primary objective is to design the vehicle so that its exterior design minimizes the consequences of a primary collision (a collision involving persons outside the vehicle and the vehicle itself).

The most severe injuries sustained by pedestrians result from impact with the front of the vehicle or the road, in addition to which the precise accident sequence greatly depends on body size. The consequences of collisions involving two-wheeled vehicles and passenger cars can

only be slightly improved by passenger-car design due to the two-wheeled vehicle's often considerable inherent energy component, its high seat position, and the wide dispersion of contact points. The design features that can be incorporated in a passenger car are, for example:

- Movable front lamps
- Recessed windshield wipers
- Recessed drip rails
- Recessed door handles
- Deformable vehicle front, including hood.

See also ECE-R26 [20], 2009/78/EG [21], 2004/90/EC [22].

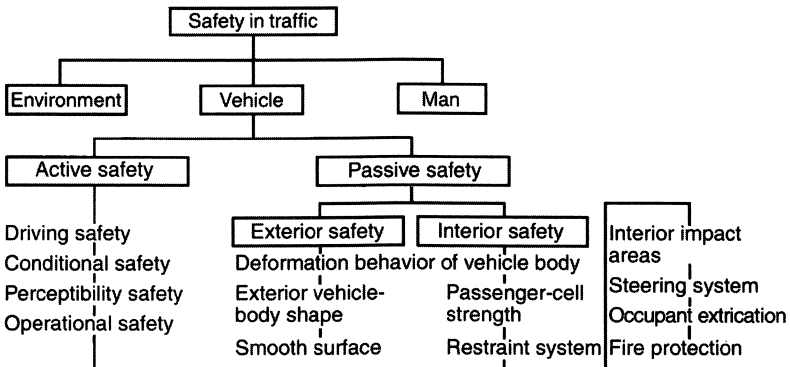
Interior safety

The term "interior safety" covers vehicle measures whose purpose is to minimize the acceleration and forces acting on vehicle occupants in the event of an accident, provide sufficient survival space, and ensure the operability of vehicle components critical to the extrication of occupants from the vehicle after the accident has occurred. The determining factors for passenger safety are:

- Deformation behavior of vehicle body,
- Passenger-cell strength, size of the survival space during and after impact,
- Restraint systems,

Figure 5: Safety in traffic

Terms and influencing factors



- Impact area in the interior (FMVSS 201 [23]),
- Steering system,
- Occupant extrication,
- Fire protection.

Laws governing inner safety (frontal and side impacts):

- Protection of vehicle occupants in the event of an accident, in particular restraint systems (FMVSS 208 [24] amended version, FMVSS 214 [25], ECE-R94 [26], ECE-R95 [27], injury criteria),
- Windshield mounting (FMVSS 212 [28]),
- Penetration of the windshield by vehicle body components (FMVSS 219 [29]),
- Storage compartment lids (FMVSS 201 [23]),
- Fuel leakage prevention (FMVSS 301 [30]).

Deformation behavior of vehicle body

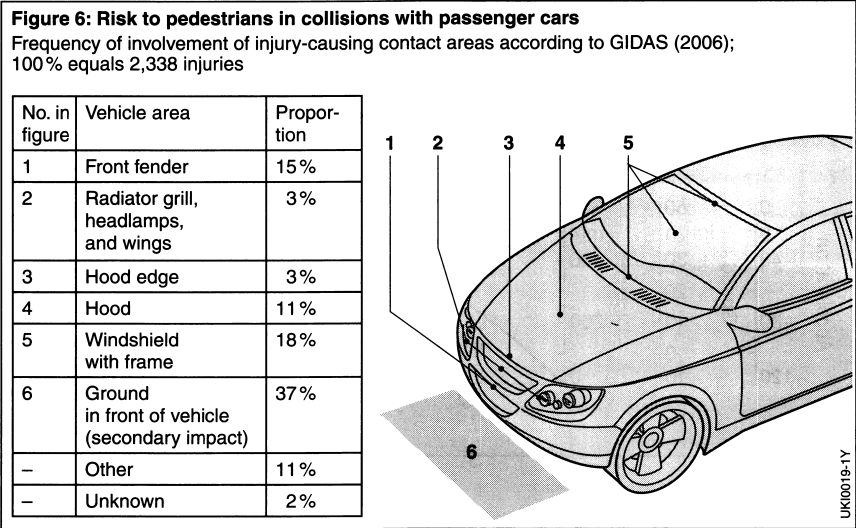
Due to the frequency of frontal collisions, an important role is played by the legal regulations for frontal impact tests in which a vehicle is driven at a speed of 30 mph (48.3 km/h) into a rigid barrier which is either perpendicular or inclined at an angle of up to 30° relative to the car's longitudinal axis. Figure 6 shows the distribution of collision types for accidents resulting in injuries to ve-

hicle occupants. Source: GIDAS, German In-Depth Accident Study (research project by BASt and FAT).

As virtually 50% of frontal collisions mainly affect only one half of the front of the vehicle, offset frontal impact with a coverage of 30 to 50% of the vehicle width is carried out worldwide.

Excerpt from ECE-R94: "The barrier must be configured in such a way that the vehicle contacts it on the driver's side first. If the test can be conducted with a vehicle with either right- or left-hand steering, it must be conducted with the least favorable type of steering which is established by the technical agency responsible for the test".

In a frontal collision, kinetic energy is absorbed through deformation of the fender, the front of the vehicle, and in severe cases, the forward section of the passenger cell (engine compartment bulkhead). Axles, wheels (rims) and the engine limit the deformable length. Adequate deformation lengths and displaceable drivetrain assemblies are necessary, however, in order to minimize passenger-cell acceleration. Depending on vehicle design (body shape, type of drive, and engine position), vehicle mass and size, a frontal impact with a barrier at approx. 50 km/h results in permanent deformation in the



forward area of 0.4 to 0.7 m. Damage to the passenger cell should be minimized. This concerns primarily:

- the engine-bulkhead area (displacement of steering system, instrument panel, pedals, constriction of footwell),
- the underbody (lowering or tilting of seats),
- the side structure (ability to open the doors after an accident).

Acceleration measurements and evaluations of high-speed films allow a precise analysis of deformation behavior. Dummies of various sizes are used to simulate vehicle occupants and provide measured data for forces acting on the head, neck, chest, and legs.

The side impact, as the next most frequent type of accident, places a high risk of injury on vehicle occupants due to the limited energy-absorption capacity of trim and structural components, and the resulting high degree of vehicle interior deformation.

The risk of injury is largely influenced by the structural strength of the side of the vehicle (pillar/door joints, top/bottom pillar points), load-carrying capacity of floorpan cross-members and seats, and the design of inside door panels (FMVSS 214 and 301, ECE-R95, Euro NCAP, and US SINCAP). Additional airbags in the doors or seats and the headliner have considerable potential for reducing the risk of injury.

In the rear impact test, deformation of the passenger cell must be minor at most. It should still be possible to open the doors, the edge of the trunk lid should not penetrate the rear window or enter the vehicle interior, and fuel-system integrity must be preserved (FMVSS 301).

Roof structures are investigated by means of rollover tests and quasi-static car-roof crush tests (FMVSS 216).

In addition, some manufacturers subject their vehicles to the inverted vehicle drop test in order to test the fatigue strength of the roof structure (survival space) under extreme conditions (the vehicle falls from a height of 0.5 m onto the left front corner of its roof).

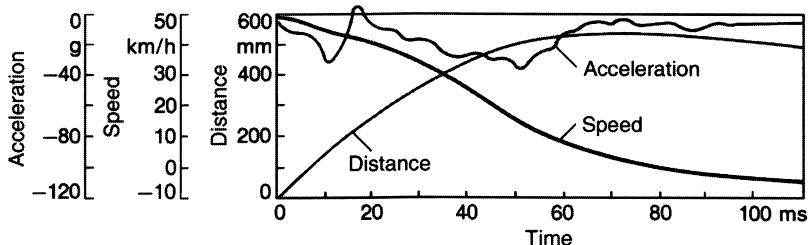
Integral safety

The overlap between active and passive safety is increasing on account of the sensor technology used by both disciplines. That is why more and more systems are being developed which precondition the occupants better for a possible accident (e.g. Pre-Safe).

References

- [1] DIN 33408-1: Body templates – Part 1: For seats of all kinds.
Addendum 1: Body templates for seats of all kinds; application examples.
- [2] SAE J 826: Devices for Use in Defining and Measuring Vehicle Seating Accommodation.

Figure 7: Acceleration, speed, and distance traveled of a passenger cell when impacting a barrier at 50 km/h



- [3] ISO 6549: Road vehicles – Procedure for H- and R-point determination.
- [4] SAE J 941: Motor Vehicle Drivers Eye Locations.
- [5] Council Directive 77/649/EEC of 27 September 1977 on the approximation of the laws of the Member States relating to the field of vision of motor vehicle drivers.
- [6] ISO 3832: Passenger cars – Luggage compartments – Method of measuring reference volume.
- [7] 49 CFR Part 581 – Bumper Standard.
- [8] Canada Motor Vehicle Safety Standard – CMVSS 215 – Bumpers.
- [9] ECE R42: Uniform provisions concerning the approval of vehicles with regard to their front and rear protective devices (bumpers, etc.).
- [10] Regulation No. 11 of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of vehicles with regard to door latches and door retention components.
- [11] FMVSS 206: Door locks and door retention components.
- [12] FMVSS 401: Interior trunk release.
- [13] FMVSS 207: Seating systems.
- [14] FMVSS 202a: Head restraints; Mandatory applicability begins on September 1, 2009.
- [15] ECE R17: Regulation No. 17 of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of vehicles with regard to the seats, their anchorages and any head restraints.
- [16] ECE R25: Regulation No. 25 of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of head restraints (headrests), whether or not incorporated in vehicle seats.
- [17] Council Directive 74/408/EEC of 22 July 1974 on the approximation of the laws of the Member States relating to the interior fittings of motor vehicles (strength of seats and of their anchorages).
- [18] Council Directive 78/932/EEC of 16 December 1978 on the approximation of the laws of the Member States relating to the head restraints of seats of motor vehicles.
- [19] FMVSS 302: Flammability of materials used in the occupant compartments of motor vehicles.
- [20] ECE R26: Regulation No. 26 of the United Nations Economic Commission for Europe (UN/ECE) – Uniform provisions concerning the approval of vehicles with regard to their external projections.
- [21] Regulation (EC) No. 78/2009 of the European Parliament and of the Council of 14 January 2009 on the type-approval of motor vehicles with regard to the protection of pedestrians and other vulnerable road users, amending Directive 2007/46/EC and repealing Directives 2003/102/EC and 2005/66/EC.
- [22] 2004/90/EC: Commission Decision of 23 December 2003 on the technical prescriptions for the implementation of Article 3 of Directive 2003/102/EC of the European Parliament and of the Council relating to the protection of pedestrians and other vulnerable road users before and in the event of a collision with a motor vehicle and amending Directive 70/156/EEC.
- [23] FMVSS 201: Occupant protection in interior impact.
- [24] FMVSS 208: Occupant crash protection.
- [25] FMVSS 214: Side impact protection.
- [26] ECE R94: Uniform provisions concerning the approval of motor vehicles with regard to the protection of the occupants in the event of a frontal collision.
- [27] ECE R95: Uniform provisions concerning the approval of motor vehicles with regard to the protection of the occupants in the event of a lateral collision.
- [28] FMVSS 212: Windshield mounting.
- [29] FMVSS 219: Windshield zone intrusion.
- [30] FMVSS 301: Fuel System Integrity.

Vehicle bodies, commercial vehicles

Classification of commercial vehicles

Commercial vehicles are used for the safe and efficient transportation of persons and freight. In this respect, the degree of economic efficiency is determined by the ratio of usable space to overall vehicle volume, and of payload to gross vehicle weight. Dimensions and weights are limited by legal regulations.

A wide variety of vehicle types meet the demands of local and long-distance transportation, as well as the demands encountered on building sites and in special applications (examples in Figure 1). Commercial vehicles can basically be subdivided into the following categories: light utility vans, medium- and heavy-duty

trucks, buses, tractor units, construction and agricultural machines, and special vehicles (e.g., piste basher, airfield fire truck).

Due to the wide diversity of vehicle types, the process of calculating the dimensions of the body structures (unitized body, cab, chassis, etc.) takes on a major importance right from the earliest stages of design. Building on experience with comparable vehicles, benchmark designs (volume-sales units, worst-case configurations) are defined by simulation and calculation with the aid of gradually refined complete-vehicle models using FEM (finite-element method) or MBS (multiple-body simulation). In this way, the rigidity, operational integrity, and vibration, acoustic and crash characteristics, etc. of relevant body-structure variants can be obtained to a substantial degree by computational means, even before testing starts. Structural calculations also take account of the requirements of (international) statutory safety standards.

Figure 1: Overview of commercial vehicles (Examples).

- a) Light utility van, b) Truck,
- c) Articulated road train,
- d) High-capacity road train,
- e) Semitrailer (Europe),
- f) Semitrailer (NAFTA), g) Bus.

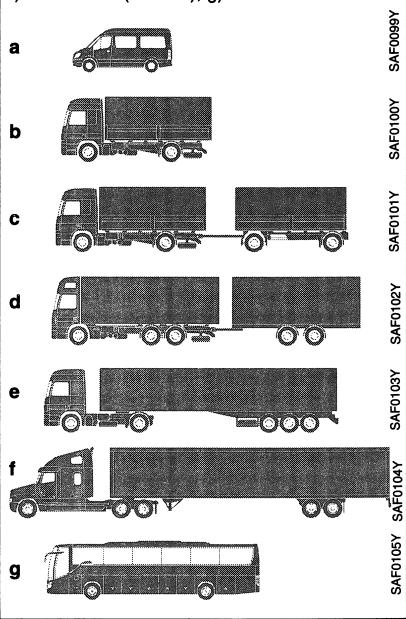
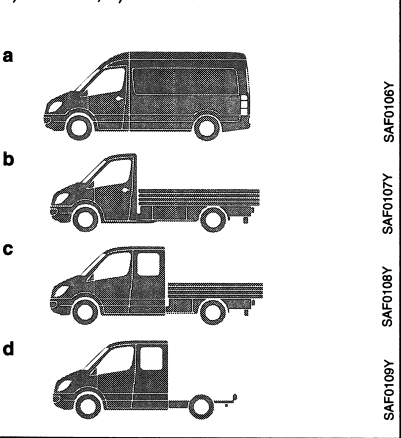


Figure 2: Overview of light utility vans (Examples).

- a) Panel van, b) Flatbed van,
- c) Twin cab, d) Chassis.



Light utility vans

Areas of application

These are light utility vans (2 to 7 t) used in the transportation of persons and in local freight distribution. Light utility vans with more powerful engines are also increasingly deployed in pan-European long-distance transport duties involving high mileages (express delivery services, overnight courier services). In both cases, stringent demands are made on the vehicle in terms of agility, performance, user-friendliness, and safety.

Body-structure variants of light utility vans

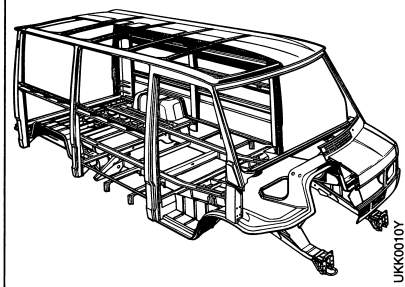
The design concepts are based on front-mounted engine, front or rear-wheel drive, independent suspension, or rigid axle and, with overall weights of over 3.5 to 4 t, twin tires on the rear axle.

The product range includes enclosed-body multipurpose panel vans and platform-body vehicles, as well as low-bed and high-bed platform trucks with special superstructures and crew cabs (examples in Figure 2).

Up to approximately 6 t overall weight, the bodies form an integral load-structure unit together with the chassis (Figure 3). The body and chassis frameworks consist of sheet-metal pressed elements and flanged profiles similar to passenger cars.

Light utility vans with platform bodies have a ladder-type frame with open or closed side members and cross-members as the primary load-bearing structure (similar to trucks, next section, Figure 5). These open designs also serve to build up box or recreational-vehicle bodies.

Figure 3: Light utility van, load-bearing unit



Medium- and heavy-duty trucks and tractor units

Body

Trucks are divided into four essential sub-types: long-distance transport vehicles, delivery vehicles, construction vehicles, and special vehicles. What all these vehicles have in common is a supporting frame structure on which a resiliently mounted cab is located and which constitutes a body which is connected with shear strength to the frame. The maximum dimensions and maximum permitted vehicle weights are subject to statutory provisions, which sometimes differ significantly from country to country. The maximum dimensions with regard to overall and semi-trailer lengths are stipulated for example in Europe by Directive 96/53/EU [1], the new version of which 2015/719/EU [2] is currently being technically elaborated. In the NAFTA zone, however, only the semi-trailer length, but not the overall length, is subject to restriction.

Equipped with a front-mounted engine, the drive is provided via one or more dual-tire rear axles. In individual cases, the rear axle is fitted with single tires. Four-wheel drive with an additional transfer case including interaxle-differential lock is also used for construction-site applications (off-road) with high traction requirements. The powered axles in this case have as standard axle-differential locks in their differentials and interaxle-differential locks for tandem-axle assemblies.

Chassis

Designations

The type of truck chassis (Figure 4) is given its designation in accordance with the convention $N \times Z/L$, where N denotes the number of wheels or wheel pairs, Z the number of driven wheels or wheel pairs, and L the number of steered wheels (twin wheels count as one wheel pair). If L is not specified (e.g., 4×2), the vehicle has two steered front wheels.

Wheel suspension

Truck chassis have pneumatic- or leaf-spring-suspension rigid front and rear axles. Independent wheel suspensions are only used occasionally. Pneumatic

suspensions reduce the body acceleration to improve driving comfort, protect the load, and reduce strain on the road. They also facilitate the raising and lowering of swap bodies and the attachment and removal of semitrailers. These applications are usually used with 4×2 and 6×2 chassis for long-distance transport and delivery traffic.

Tandem-axle assemblies

For construction-site use or in the case of increased traction requirements the chassis are equipped with several powered axles (e.g., 4×4, 6×4, 6×6, 8×4, 8×8). In this case, the rear axles are combined to form a tandem-axle assembly. For the most part a steel-spring suspension is used, but pneumatic suspensions are also featured on the rear axles for road-based applications (e.g., building-material or low-loader transport) with the previously mentioned advantages. The axle

load in a tandem-axle assembly is compensated in the case of the steel spring mechanically by pivoted mounting (center mounting) between the axles. In the case of non-combined pneumatically sprung single axles, the axle load is normally compensated pneumatically by altering the spring stiffness of the pneumatic suspension of the single axles.

Leading and trailing axles

Three-axle vehicles (6×2) are fitted with either a leading or a trailing axle (nonpowered axles in front of or behind the driven axle) to increase the payload capacity. If the chassis is fitted with a leading axle, this is mounted as an additional axle in front of the powered rear axle and provided with pneumatic suspension.

These leading axles usually have a steered design in order to avoid constraints when cornering, deliver a small turning circle, and thereby increase the maneuverability of the complete vehicle. The steering was effected in the first designs by a steering linkage branched from the steering axle, but is now equipped as a rule with electro-hydraulic steering because this does not require a linkage that takes up valuable space.

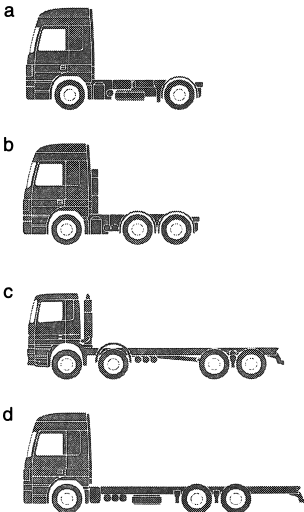
It is possible to equip the leading axle with a smaller wheel/tire size differing from the powered axle, but in this case the bearing loads of the leading axle are lower (4 to 5 t instead of a possible 7.5 to 9 t). There is also the combination of the leading axle with a lifting device. Then, when the vehicle is driving empty or with a low load, the axle is lifted from the driving position via a further air-spring bellows operating in the axially centered direction. This decreases rolling resistance and consequently reduces fuel consumption and also tire wear. Both these factors help to improve the vehicle's economic efficiency.

In the case of chassis designs with a trailing axle, this axle is mounted behind the powered rear axle. It is as a rule provided in on-road vehicles with pneumatic suspension and fitted with single tires, and in this case usually has a bearing load of 7.5 t. This variant can also be steered. Where trailing axles were still rigid when they began to be used in the late 1960s with the emergence of swap-body artic-

Figure 4: Truck chassis

(Examples).

- a) 4×2 (four wheels, two of which driven),
- b) 6×2/4 (six wheels, two of which driven, four steered),
- c) 8×6/4 (eight wheels, six of which driven, four steered),
- d) 6×2 (six wheels, two of which driven).



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ulated road trains, steered trailing axles came to the fore with the desire for improved maneuverability against the back-drop of the increased use of three-axle chassis in delivery traffic. In view of the space conditions the trailing axle here is as a rule fitted with electro-hydraulic steering so that these variants can be flexibly used with different chassis and wheelbases. The use of a steered trailing axle offers, with a corresponding vehicle configuration, the combination of the load capacity of a three-axle chassis with the maneuverability of two-axle vehicle, which affords considerable benefits in economic terms. The trailing axles are also designed to be liftable virtually across the board with the previously mentioned advantages.

Where 6×2 chassis are operated on dirt roads or outside Europe, there are twin-tire rigid trailing axles with a bearing load of 9 to 10 t. As a rule, these are not designed to be liftable, but liftable variants are possible.

For special chassis the multiple fitment of leading and trailing axles or even the use of powered leading or trailing axles is conceivable. These chassis variants are conceived as special constructions and constitute technical challenges in that each axle must be assigned a different steering angle for the wheels so that the

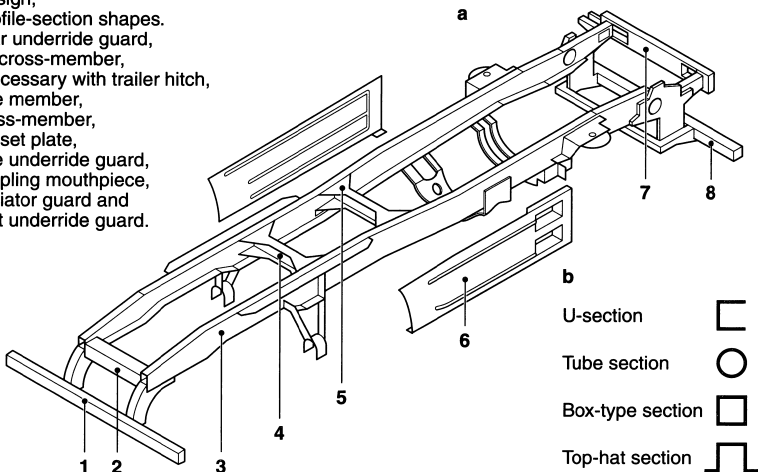
vehicle runs without excessive tire wear when cornering and complies with the turning-circle requirements. Combining a number of leading and trailing axles can thus also produce five- to eight-axle chassis which make possible gross vehicle weight ratings of over 60 t, for example for the mounting of concrete pumps or lifting work platforms with a working height of up to 112 m.

Chassis frame

The chassis frame is the commercial vehicle's actual load-bearing element. It is designed as a ladder-type frame, consisting of side members and cross-members (Figure 5). The vehicle body is bolted onto and the driver's cab is positioned on the chassis frame (Figure 6). The dimensions of the members are chosen to suit the required severity of application and load capacity (light- and heavy-duty commercial vehicles), but also in line with cost and weight considerations. The choice of profile sections (number and thickness) determines the level of torsional stiffness. Torsionally flexible frames are preferred in medium- and heavy-duty trucks because they allow the suspension to cope better with uneven terrain. Torsionally stiff frames are more suitable for light-duty delivery vehicles.

Figure 5: Ladder-type frame for a truck

- a) Design,
b) Profile-section shapes.
- 1 Rear underride guard,
 - 2 Tail cross-member,
 - if necessary with trailer hitch,
 - 3 Side member,
 - 4 Cross-member,
 - 5 Gusset plate,
 - 6 Side underride guard,
 - 7 Coupling mouthpiece,
 - 8 Radiator guard and front underride guard.

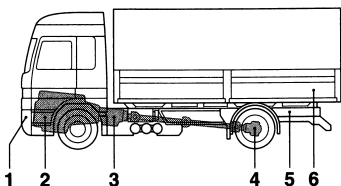


Apart from the force introduction points, critical points in the chassis-frame design are the side- and cross-member junctions (Figure 7). Special gusset plates or pressed cross-member sections form a broad connection basis. The junctions are riveted, bolted, and welded. U- or L-shaped side-member inserts provide increased frame flexural strength and reinforcement at specific points.

The chassis frame also serves to accommodate by means of a variety of holders a wide range of add-on components such as fuel tanks, battery-device holders, compressed-air tanks, exhaust-gas system, or spare wheel. The configuration varies here, depending on the application profile required. Special mounted installations such as, for example, loading cranes or platform lifts are likewise mounted on the chassis frame for corresponding applications.

Figure 6: Truck assemblies

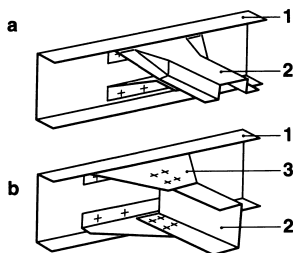
1 Cab, 2 Engine, 3 Transmission, 4 Axle, 5 Chassis frame, 6 Body.



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Figure 7: Junctions

a) Cap cross-member, b) U-cross-member.
1 Side member, 2 Cross-member, 3 Gusset plate.



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Driver's cab

There are a variety of cab designs available depending on the vehicle concept. In delivery vehicles and for public-service use, low, convenient entrances are an advantage, whereas in long-distance transport applications, space and comfort for example with a level floor are more important. Modular design concepts allow for short, medium and long cab versions while retaining the same front, rear and doors.

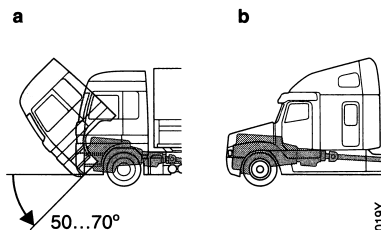
The cab is connected to the chassis frame by the cab mounting. A distinction is made here between comfort and standard mountings with different spring-and-damper combinations or transverse leaf-spring links with natural frequencies of 1 to 6 Hz.

From the design-concept viewpoint, a distinction must be made between cab-over-engine (COE) and cab-behind-engine (CBE) vehicles (Figure 8). In the case of cab-over-engine (COE) vehicles, the bulkhead and steering system are positioned right at the front of the vehicle. The engine is located under the cab (in the case of a level floor) or underneath an engine tunnel between the driver and the co-driver. The entrance is positioned in front of or above the front axle. A mechanical (by pretensioned torsion bars) or hydraulic cab-tipping mechanism provides access to the engine.

In cab-behind-engine (CBE) vehicles, the engine/transmission assembly is mounted ahead of the cab firewall under

Figure 8: Driver's cab

a) Cab over engine,
b) Cab behind engine.



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a steel or plastic hood which is usually tiltable for reasons of accessibility. The driver enters the cab behind the front axle.

The standards required of the driver's cab with regard to aerodynamics, choice of materials, corrosion or equipment must be viewed as the same as the standards required of an automobile body. This means that, to save fuel and consequently CO₂ emissions, the aerodynamic drag of firstly the cab and secondly the entire truck-trailer must be reduced. Both the tractor vehicle and the trailer or the semitrailer must be included in the analyses for this purpose. Field tests show that including the semitrailer in the optimization of the complete-vehicle aerodynamics can achieve fuel savings of between 2 % and 4.5 %.

Body structures

Specific body structures such as flatbeds, box vans, dump-truck deep-beds, tankers, and concrete mixers permit the economical and efficient transportation of a wide variety of freight and materials. The body and load-bearing chassis frame are joined sometimes by means of auxiliary frames with non-positive or positive attachments. Special design features (e.g., sprung mountings in the forward body area) are required to connect the chassis frame (which usually has low torsional strength) to a rigid body (e.g., box-type). For off-road vehicles the torsional flexibility of

the chassis frame must not be limited by a rigid body structure. Three-point mountings of the body structure are therefore used. For these vehicles coil instead of leaf springs are also used in the chassis to achieve large spring deflections.

Articulated road trains and semitrailer rigs are used in long-distance transport (Figure 1). As the size of the transportation unit increases, the costs relative to the freight volume decrease. Load volume is increased by reducing the empty spaces between the cab, body, and trailer (high-capacity road train, Figure 1d). Advantages of semitrailer rigs lie in the greater uninterrupted loading length of the cargo area and possible shorter inoperative times of the tractor units. Measures to improve aerodynamics, such as front and side panels on the vehicle and specially adapted air deflectors from the cab to the body, are applied to minimize fuel consumption.

The use of longer heavier vehicles which do not exceed the gross weight rating of 40 t but extend the overall road-train length to 25.25 m (instead of 16.50 m for semitrailers and 18.75 m for articulated road trains) can generate additional economic benefits. These longer heavier vehicles are used for light but bulky transported goods, e.g., for molded parts for the automotive industry, or in the transport of consumer goods.

Buses

The bus market offers a specific vehicle for practically every application. This results in a wide range of bus types, which differ in their overall dimensions (length, height, width) and appointments (depending on the application) (Figure 9).

Bus types

Microbuses

Minibuses carry up to approx. 20 passengers. These vehicles have been developed from light utility vans weighing up to approximately 4.5 t.

Mini- and midibuses

Buses which can carry up to approximately 25 persons are called mini- or midibuses. The transition from minibus to midibus is fluid. These vehicles have for the most part been developed from light utility vans weighing up to approximately 7.5 t. They are occasionally built on ladder-type chassis of light-duty trucks, or a unitized integral structure is used. A modified suspension design and special measures carried out on the body (e.g., flexible mountings) result in optimum ride comfort and low noise levels.

City buses

City buses are equipped with seating and standing rooms for scheduled routes. The short intervals between stops in suburban passenger transportation operations require rapid passenger turnover. This is achieved by wide doors that open and close swiftly, low boarding heights (approx. 320 mm) and low floor heights (approx. 370 mm).

Main specifications for a standard municipal-service bus:

- Vehicle length approx. 12 m,
- Permissible total weight 18.0 t,
- Number of seats 32 to 44,
- Total passenger capacity approx. 105 persons.

The use of double-decker buses (length 12 m carrying up to approximately 130 passengers), three-axle rigid buses (length up to 15 m carrying up to approximately 135 passengers), and three-axle articulated buses (length up to approxi-

mately 18 m carrying 160 to 190 passengers) provides increased transport capacity.

Now articulated buses with an overall length of almost 21 m and a total weight of 32 t are used in large numbers as city buses. These make use of an additional steered trailing axle which increases the bus' weight capacity and at the same time improves the maneuverability despite the greater overall length.

Intercity/overland buses

Depending on the application (standing passengers are not permitted at speeds over 60 km/h), low-floor designs featuring low boarding and floor heights (as in city buses) or higher floors and small luggage compartments (already very much like tour buses) are used. Intercity buses come in lengths of 11 to 15 m as rigid vehicles or in lengths of 18 m as articulated vehicles.

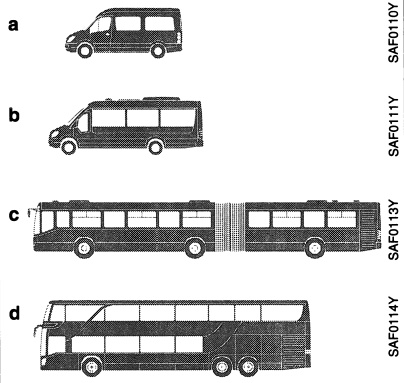
Tour buses (coaches)

Tour buses are designed to provide comfortable travel over medium and long distances. They range from the low, two-axle standard bus through to the double-decker luxury bus. Tour buses come in lengths of 10 to 15 m. Articulated buses were only occasionally used in intercity-bus applica-

Figure 9: Overview of buses

(Examples).

- a) Microbus,
- b) Mini- and midibus,
- c) City bus,
- d) Tour bus.



tions and did not gain acceptance. Double-deckers, on the other hand, are frequently used as intercity buses, particularly on long-distance bus routes which are operated with tour buses.

Trailer operation

Buses with trailers are increasingly being used. This applies to all applications, i.e., city and intercity/overland buses, as well as to tour buses.

In city-bus applications trailers are specially approved to carry passengers and at peak times are hitched to the towing vehicle to meet passenger demand. This type was used up until the 1960s, but disappeared very quickly with the emergence of articulated buses. Since around 2010 the city-bus/trailer combination has been back in favor so as to provide a flexible response to passenger fluctuations depending on the time of day on the outskirts of big cities.

With intercity/overland buses trailers serve to transport luggage or to carry bicycles in vacation regions so as to be able to offer an attractive and flexible means of transport for changing to local public transport.

In the case of tour buses, particularly double-deckers, which in view of their design can offer minimal luggage space, the luggage trailer is a good solution as an economical and comfortable means of travel. The approval of trailers for cruising speeds identical to buses without trailers (max. 100 km/h) entails no drawbacks in terms of travel times, increasing the attractiveness.

Body

Lightweight design based on a unitized body has gained acceptance as standard (integral construction). The body and base frames, which are firmly welded together, consist of pressed grid-type support elements and rectangular tubes (Figure 10).

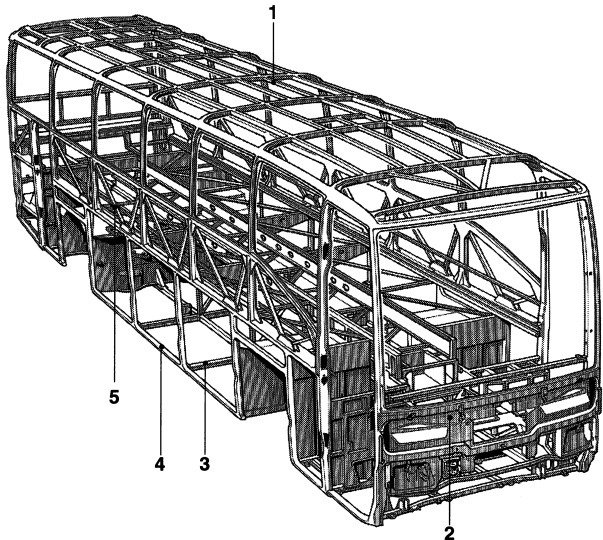
In the case of the chassis structure, the body is positioned on a supporting ladder-type frame (similar to trucks). Except for mini- and midibuses, this design is not commonly encountered in Europe.

Chassis systems

The vertically or horizontally mounted engine drives the rear axle. Here the engine together with the transmission is not – in contrast to trucks – installed at the front

Figure 10: Unitized bus body

- 1 Body framework,
- 2 Pressed parts,
- 3 Base frame,
- 4 Rectangular tube,
- 5 Grid-type support elements.



and then connected with a long propeller shaft to the powered axle, but instead the engine/transmission unit is installed at the rear end of the bus. This also applies to articulated buses, in which the last axle is powered. Pneumatic suspension on all axles permits ride-level stabilization and a high degree of ride comfort. Intercity buses and tour buses are mainly equipped with independent suspension on their front axles. Disk brakes, frequently supported by retarders, are used on all axles.

Alternative drives

In view of the increasing sensitivity of users and operators of commercial vehicle, increasing endeavors are being made to produce alternative drives for commercial vehicles. The advantage of vehicles with alternative drives is that they comply with CO₂ legislation, avoid break-in restrictions, and satisfy the demands of society in general. These demands on the one hand relate to eliminating the dependence on petroleum fuels, whose days as fuels are numbered. They relate on the other hand to reducing the emission of CO₂ as a combustion product that contributes substantially to the greenhouse effect.

Both light utility vans and buses can be equipped for urban operation with a natural-gas, a battery-electric, or a fuel-cell drive. Likewise, battery drives are also available for heavy trucks in local goods and delivery traffic. A future development task is to come up with the best complete-vehicle concept for an electric truck and in so doing bring into line the general design of the battery with regard firstly to range, installation space, weight, and costs, and secondly to the demands of daily operation (charging cycle, charging time, useful load, ease of use).

Active and passive safety in commercial vehicles

Active safety comprises all the systems and measures which help to avoid an accident. Passive safety is intended to limit the consequences of accidents and to protect other road users. Systematic recording of accidents, accident tests with complete vehicles, and intensive computer optimization help to devise safety measures.

Active safety

A minimum of active safety systems is re-defined on a regular basis by legislative bodies. Where the antilock braking system (ABS) was developed to the stage of series production for trucks back in 1980, it gained acceptance in all trucks at a later stage as the result of appropriate legal requirements. This also applies to driving-dynamics control (Electronic Stability Control, ESC) and the active brake assistant.

The availability of a comprehensive package of active-safety systems is a necessary precondition for the application of high automated or autonomous driving, a epoch-making change in truck operation.

Passive safety

Requirements

Generally, the effectiveness and strength of occupant restraint systems has to be demonstrated. Therefore, the dimensioning of commercial-vehicle body structures must take account of aspects, such as the strength and rigidity of seat-belt anchor points on the seats and of the related body structures (seat rails, floor, frame, etc.).

In the event of a collision, the driver's cab and the passenger cabin must maintain the amount of room necessary for occupant survival, while at the same time deceleration must not be excessive. Depending on vehicle design, there are a variety of solutions to this problem. In light utility vans, front-section design is energy-absorbing as in passenger cars. In spite of shorter deformation paths and higher levels of released energy, the physiologically permissible limits are not exceeded in virtually all passenger-car

crash-test standards (legal requirements and rating tests). In light utility vans, there must also be features which prevent injury to occupants by uncontrolled movement of the payload. The static and dynamic strength of such features (partitions, cages and nets, securing eyes) has to be demonstrated mathematically or by testing.

In the case of trucks, the side members extend up to the front fender and can absorb high linear forces. Such passive-safety measures are based on accident analyses and are intended to improve the structural design of the cab. Static and dynamic stress and impact tests at the front and rear of the cab, as well as on its roof, simulate the stresses involved in a frontal impact and in accidents in which the vehicle overturns, rolls over, or in which the cargo moves. They are described in the Regulation ECE R29 [3], but the passing of these tests as a condition of type approval is only required in a few European countries.

Statistical analyses have proved that the bus is the safest means of passenger transportation. Static roof-load tests and dynamic overturning tests provide evidence of body strength. The use of flame-retardant and self-extinguishing materials for the interior equipment of the vehicle minimizes the risk of fire.

As road traffic involves many different kinds of vehicles, collisions between light and heavy vehicles cannot be avoided 100 % without the use of active-safety systems or the communication of vehicles with each other (V2V, vehicle to vehicle; V2X, vehicle to infrastructure). As a result

of the differences in vehicle weight, and incompatibility in terms of vehicle geometry and structural stiffness, the risk of injury in the lighter vehicle is greater.

The change in speed Δv during a central plastic impact for front-end or rear-end collisions between two vehicles (vehicle 1 and vehicle 2) is defined as

$$\mu = \frac{m_2}{m_1} : \Delta v_1 = \frac{\mu v_r}{1 + \mu}, \Delta v_2 = \frac{v_r}{1 + \mu},$$

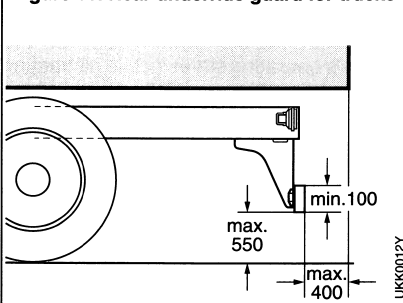
with the masses m_1 and m_2 of the vehicles involved and the relative speed v_r prior to the impact.

Side, front and rear underride guards help to reduce the danger of the lighter vehicle driving under the heavier vehicle. In other words, they serve to protect other road users (Figure 11).

References

- [1] 96/53/EU: Council Directive 96/53/EC of 25 July 1996 laying down for certain road vehicles circulating within the Community the maximum authorised dimensions in national and international traffic and the maximum authorised weights in international traffic.
- [2] 2015/719/EU: Directive (EU) 2015/719 of the European Parliament and of the Council of 29 April 2015 amending Council Directive 96/53/EC laying down for certain road vehicles circulating within the Community the maximum authorised dimensions in national and international traffic and the maximum authorised weights in international traffic (Text with EEA relevance).
- [2] ECE R 29: Regulation No. 29; Uniform provisions concerning the approval of vehicles with regard to the protection of the occupants of the cab of a commercial vehicle: Revision 1.
- [4] E. Hoepke, S. Breuer and others: *Nutzfahrzeugtechnik*. Verlag Springer Vieweg, 8th Ed., 2016.

Figure 11: Rear underride guard for trucks



Lighting equipment

Functions

Ever since the invention of the automobile, it has been accompanied by vehicle lighting. Initially candles, then petroleum and carbide lamps were used to provide lighting; today, these lamps would be classed as position or marker lamps. Only with the introduction of automotive electrics in the form of the Bosch alternator in the Adler vehicle (1913) were manufacturers able to introduce systems which generated good ranges and lived up to being called "headlamps". Further significant milestones have been

- the introduction of the asymmetrical low-beam pattern, characterized (RHD traffic) by an extended visual range along the right side of the road (1957),
- the introduction of new headlamp systems featuring complex geometrical configurations (PES, Poly-Ellipsoid System, free-form surfaces, faceted reflectors) offering efficiency-level improvements of up to 50 % (1985),
- the "Litronic" headlamp system with gas-discharge lamps (xenon lamps with luminous arc) which supplied more than twice the light generated by comparable halogen units (1990),
- adaptive front-lighting systems (AFS) with moving, dynamic PES modules (Poly-Ellipsoid System) or static, activated reflectors for turning (2003).

Lighting at the vehicle front end

The primary function of the headlamps at the vehicle front end is to illuminate the roadway so that the driver can detect traffic conditions and recognize any obstacles and hazards in good time. They also serve to identify and mark out the vehicle to oncoming traffic. The turn-signal lamps serve to show the driver's intention to change direction or to indicate a hazardous situation.

The headlamps and lights at the front end include the following:

- Low-beam headlamps
- High-beam headlamps
- Fog lamps
- Auxiliary driving lamps
- Turn-signal (direction-indicator) lamps
- Parking lamps
- Position/clearance lamps (for wide vehicles)
- Daytime running lamps (if required by law in individual countries).

Lighting at the vehicle rear end

Lights are turned on at the vehicle's rear end in accordance with the lighting and weather conditions and indicate the vehicle's position. They also indicate how the vehicle is moving and in which direction, e.g. whether the brakes are applied or the driver is intending to change direction, or whether a hazardous situation exists. The reversing lamps illuminate the roadway while the vehicle is reversing.

The lamps and lights at the rear end include the following:

- Stop lamps,
- Tail lamps,
- Rear fog warning lamps,
- Turn-signal (direction-indicator) lamps,
- Parking lamps,
- Clearance lamps (for wide vehicles),
- Reversing (backup) lamps,
- License-plate lamps.

Lighting in the vehicle interior

In the vehicle interior, priority over all other functions is given to the ease and reliability with which the switch elements can be reached and operated, and to provide the driver with sufficient information on the vehicle's operating states (while distracting him/her from driving as little as possible). These priorities dictate effectively illuminated instrument panels and discrete lighting for various control clusters (such as the sound and navigation systems), where they satisfy a prime requirement for relaxed and safe vehicle operation. Optical and acoustic signals must be prioritized according to their urgency and then relayed to the driver.

Regulations and equipment

Approval codes and symbols, Europe/ECE

Automotive lighting equipment is governed by national and international design and operating regulations, according to which the equipment in question must be manufactured and tested. For every type of lighting equipment, there is a special approval code and symbol which has to be legibly displayed on the equipment concerned. The preferred locations for approval codes and symbols are places that are directly visible when the hood is open, such as the lenses of headlamps and other lights, and the headlamp unit components. This also applies to approved replacement headlamps and lights.

If an item of equipment carries such an approval code/symbol, it has been tested by a technical inspectorate (e.g. in Germany the Lighting Technology Institute of Karlsruhe University) and approved by a licensing authority (in Germany the Federal Road Transport Office). All volume-production units which carry the approval code/symbol must conform in all respects to the type-approved unit. Examples of approval symbols:

(E1) ECE approval mark,

[e1] EU approval mark.

The number 1 following each letter indicates the type-approval test carried out and the award of approval according to ECE (Economic Commission for Europe) Regulations for Germany with Europe-wide recognition. In Europe, installation of all automotive lighting and visual signaling equipment is governed not only by national guidelines but also by the higher European directives (ECE: whole of Europe, EU, New Zealand, Australia, South Africa and Japan). In the course of the ongoing union of Europe, the implementation regulations are being increasingly simplified by the harmonization of directives and legislation.

Right-hand-drive or left-hand-drive traffic

The ECE Regulations apply by analogy to driving on the right or left. The technical requirements for lighting are mirrored around the central perpendicular of the test screen (see Figure 3). According to the Vienna Global Treaty of 1968, when traveling in countries which drive on the other side of the road, all road users are obliged to adopt measures to prevent increased glare of oncoming traffic at night due to the asymmetrical light pattern. This can be achieved either by self-adhesive strips obtainable from the vehicle manufacturer or two-way switches in the headlamps (in the case of PES).

Regulations for the USA

In the USA, lighting equipment is governed by regulations that are very different from those in Europe. The principle of self-certification compels each manufacturer as an importer of lighting equipment to ensure, and in an emergency to furnish, proof that its products conform 100% with the regulations of FMVSS 108 [27] (Federal Motor Vehicle Safety Standard) laid down in the Federal Register. There is therefore no type approval in the USA. The regulations of FMVSS 108 are partly based on the SAE industry standard (Society of Automotive Engineers).

Upgrading and conversion

Vehicles imported to Europe from other regions must be modified to comply with European directives. This applies in particular to the lighting equipment. Identical components available for the European market can be used as direct replacements. Other solutions, such as retail products or, in certain cases, retention of the original equipment, require an engineer's report. In Germany, Article 22a of the StVZO (Road Traffic Licensing Regulations) [1] requires "Approximation Certificates" for lighting equipment. Such certificates are issued by the Lighting Technology Institute of Karlsruhe University.

Subsequent alterations to type-approved headlamps and sockets invalidate the type approval and consequently the general operating license of the vehicle.

Light sources

With regard to automotive light sources, a distinction is made between thermoluminescent radiators (thermal radiators) and electroluminescent radiators. The outer electron shells of atoms of certain materials can absorb varying levels of energy by excitation (energy input). The transition from higher to lower levels may lead to the emission of electromagnetic radiation.

Thermal radiators

In the case of this type of light source, the energy level of the crystal system is increased by adding heat energy. Emission is continuous across a broad wavelength range. The total radiated power is proportional to the power of 4 of the absolute temperature (Stefan-Boltzmann law). The distribution-curve maximum is displaced to shorter wavelengths as temperature increases (Wien's displacement law) [38].

Filament lamps

Filament lamps, with tungsten filament (fusion temperature 3,660 K), are also thermal radiators. The evaporation of the tungsten and the resulting blackening of the bulb restrict the service life of this type of lamp.

Halogen lamps

A halogen filling (iodine or bromine) in the lamp allows the filament temperature to rise to close to the melting point of the tungsten. Close to the hot bulb wall, the evaporated tungsten combines with the filler gas to form tungsten halide. This is gaseous, light-transmitting and stable within a temperature range of 500 K to 1,700 K. It reaches the filament by means of convection, decomposes as a result of the high filament temperature, and forms an even tungsten deposit on the filament. In order to maintain this cycle, an external bulb temperature of approx. 300°C is necessary. To achieve this, the bulb, made of fused silica (quartz), must surround the filament closely. A further advantage of this measure is that a higher filling pressure can be used, thereby providing additional resistance to tungsten evaporation.

Gas-discharge lamps

Gas-discharge lamps are electroluminescent radiators and distinguished by their higher luminous efficiency. A gas discharge is maintained in an enclosed, gas-filled bulb by applying a voltage between two electrodes. The atoms of the emitted gas are excited by collisions between electrons and gas atoms. The atoms excited in the process give off their energy in the form of luminous radiation.

Examples of gas-discharge lamps are sodium-vapor lamps (street lighting), fluorescent lamps (interior illumination) and D lamps for motor-vehicle applications (Litronic).

Light-emitting diodes

A light-emitting diode or LED is an electroluminescent lamp. Thanks to their robustness, high energy efficiency, fast reaction time, and compact construction, LEDs are used as illuminators or displays in a wide range of applications. They are used in motor vehicles in the passenger compartment to provide illumination, as displays or for display backlighting. In the exterior areas LEDs are fitted in particular in auxiliary stop lamps and tail lamps and, with further increases in their luminous efficiency, can increasingly be used in lamps and main functions at the front end of the vehicle.

Motor-vehicle bulbs

Replaceable filament bulbs for motor-vehicle lighting must be type-approved in accordance with ECE R37 [10], replaceable gas-discharge light sources in accordance with ECE R99 [19]. Other light sources which do not comply with these regulations (LEDs, neon tubes, special bulbs) are permitted, but can only be installed as a fixed component part of a lamp or as a "light-source module".

Bulbs complying with ECE R37 are generally available in 12 V versions, some bulbs also in 6 V and 24 V versions (Table 1). To help avoid mix-ups, different bulb types are identified by different base shapes. Bulbs of differing operating voltages are labeled with this voltage in order to avoid mix-ups in the case of identical bases. The bulb type suitable in each case must be indicated on the equipment.

A voltage increase in halogen lamps of 10% results in a 70% reduction in service life and a 30% increase in luminous flux (Figure 1, [39]).

The luminous efficiency (lumens per watt) represents the bulb's photometrical efficiency relative to its power input. The luminous efficiency of vacuum bulbs is 10 to 18 lm/W. The higher luminous efficiency of halogen lamps (22 to 26 lm/W) is primarily a consequence of increasing the filament temperature. Gas-discharge bulbs provide a luminous efficiency in the order of 85 lm/W for substantial improvements in low-beam performance.

LEDs today achieve a luminous efficiency in the range of 50 lm/W (LEDs with high power consumption) or 100 lm/W (LEDs with low power consumption). An increase in luminous efficiency of up to 25% is expected in the next few years.

Main-light functions

Low beam (dipped beam)

The main light for driving at night is provided by low-beam headlamps. The creation of the characteristic light-dark boundary was one of the technological milestones in lighting technology.

The "dark above/bright below" distribution pattern resulting from the light-dark boundary furnishes acceptable visual ranges under all driving conditions. This configuration reduces glare, to which approaching traffic is exposed, within reasonable limits, and at the same time it supplies relatively high luminous intensity in the area below the light-dark boundary.

The light distribution pattern must combine maximum visual ranges with minimum glare effect. These demands are supplemented by other requirements affecting the area directly in front of the vehicle. For instance, the headlamps must provide assistance when cornering, i.e. the light distribution pattern must extend beyond the left and right-side extremities of the road surface.

High beam

High-beam headlamps illuminate the road to the maximum range. This creates a high luminous intensity, depending on the distance, acting on all objects in the space reserved for traffic. High-beam headlamps are therefore only permitted when they do not dazzle oncoming traffic (glare).

The high traffic density on modern roads severely restricts the use of high-beam headlamps.

Installation and regulations

Designs

Global regulations mandate two headlamps for low beam and at least two (or the option of four) high-beam units for all dual-track vehicles. The light color is white.

Dual-headlamp system

The dual-headlamp system (Figure 2a) uses lamps with two light sources (halogen double-filament bulbs (H4), US sealed beam) for high and low beams using shared reflectors. In headlamps with gas-

Figure 1: Influence of operating voltage on some data of halogen lamps

(Source: [39])

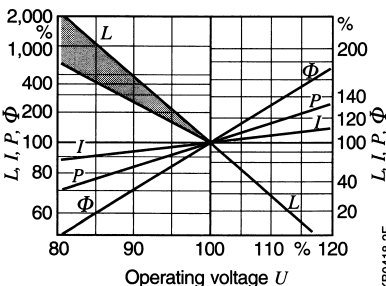
L Service life (the dispersion width during operation with undervoltage is necessitated by the presence of halogen),

U Operating voltage,

I Lamp current,

P Lamp power output,

Φ Luminous flux.



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Table 1: Specifications for the main motor-vehicle bulbs (not including motorcycle bulbs)

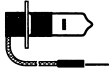
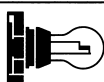
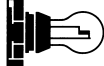
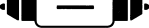
Application	Cate- gory	Voltage Nominal values V	Power Nominal values W	Luminous flux Reference values Lumen	Base type IEC	Illustration
Fog lamps, high beam, low beam in 4 HL	H1	6 12 24	55 55 70	1,350 ²⁾ 1,550 1,900	P 14.5 e	
Fog lights, high beam	H3	6 12 24	55 55 70	1,050 ²⁾ 1,450 1,750	PK 22s	
High beam, low beam	H4	12 24	60/55 75/70	1,650/ 1,000 ¹⁾ , ²⁾ 1,900/1,200	P 43 t - 38	
High beam, low beam (dipped beam) in 4 HL, fog lights	H7	12 24	55 70	1,500 ²⁾ 1,750	PX 26 d	
Fog lights, static cornering headlamps	H8	12	35	800	PGJ 19-1	
High beam	H9	12	65	2,100	PGJ 19-5	
Low beam, fog lights	H11	12 24	55 70	1,350 1,600	PGJ 19-2	
Fog lights	H10	12	42	850	PY 20 d	
High beam, daytime running lamps	H15	12 24	55/15 60/20	260/1,350 300/1,500	PGJ 23t-1	
Low beam (dipped beam) in 4-HL systems	HB4	12	51	1,095	P 22 d	
High beam in 4-HL systems	HB3	12	60	1,860	P 20 d	
Low beam, high beam	D1S	85 12 ⁵⁾	35 approx. 40 ⁵⁾	3,200	PK 32 d-2	
Low beam, high beam	D2S	85 12 ⁵⁾	35 approx. 40 ⁵⁾	3,200	P 32 d-2	
Low beam, high beam	D2R	85 12 ⁵⁾	35 approx. 40 ⁵⁾	2,800	P 32 d-3	

Table 1 (contd.): Specifications for the main motor-vehicle bulbs (not including motorcycle bulbs)

Application	Category	Voltage Nominal values V	Power Nominal values W	Luminous flux Reference values Lumen	Base type IEC	Illustration
Stop lamps, turn-signal lamps, rear fog light, backup lamps	P 21 W PY 21 W ⁶⁾	6, 12, 24	21	460 ³⁾	BA 15 s	
Brake lamp/ tail lamp	P 21/5 W	6 12 24	21/5 ⁴⁾ 21/5 21/5	440/35 ³⁾ , ⁴⁾ 440/35 ³⁾ , ⁴⁾ 440/40 ³⁾	BAY 15d	
Position lamp, tail lamp	R 5 W	6 12 24	5	50 ³⁾	BA 15 s	
Tail lamp	R 10 W	6 12 24	10	125 ³⁾	BA 15 s	
Daytime running lamps	P 13 W	12	13	250 ³⁾	PG 18.5 d	
Brake lamp, turn signal	P 19 W PY 19 W	12 12	19 19	350 ³⁾ 215 ³⁾	PGU 20/1 PGU 20/2	
Rear fog light, backup lamp, front turn signal	P 24 W PY 24 W	12 12	24 24	500 ³⁾ 300 ³⁾	PGU 20/3 PGU 20/4	
Stop, turn-signal, rear fog, backup lamps	P 27 W	12	27	475 ³⁾	W 2.5 x 16 d	
Brake lamp/ tail lamp	P 27/7 W	12	27/7	475/36 ³⁾	W 2.5 x 16 q	
License-plate lighting, tail lamp	C 5 W	6 12 24	5	45 ³⁾	SV 8.5	
Position lamp	H 6 W	12	6	125	BAX 9 s	
Position lamp, license-plate lighting	W 5 W	6 12 24	5	50 ³⁾	W 2.1 x 9.5 d	
Position lamp, license-plate lighting	W 3 W	6 12 24	3	22 ³⁾	W 2.1 x 9.5 d	

¹⁾ High/low beam. ²⁾ Specifications at test voltage 6.3; 13.2 or 28.0 V.

³⁾ Specifications at test voltage of 6.75; 13.5 or 28.0 V. ⁴⁾ Main/secondary filament.

⁵⁾ With ballast unit. ⁶⁾ Yellow-light version.

discharge bulbs the dual function is achieved by focusing or defocusing the xenon burner in a shared reflector. In bi-xenon projection systems a screen is moved out of or into the beam path.

Quad-headlamp system

Two of the headlamps in a quad-headlamp system provide both high and low beam, while the second pair only provides high-beam illumination (Figure 2b). The light functions of projection and reflection systems can be used in any combination. The low-beam headlamps can additionally be combined with the fog lamps (Figure 2c).

Important definitions of device design

Grouped design

A single housing, but with different lenses and bulbs. Example:

- Multiple-compartment rear-lamp assemblies containing different individual light units.

Combined design

A single housing and bulb assembly with more than one lens. Example:

- Combined tail lamp and license-plate lamp.

Nested design

Common housing and lens, but with individual bulbs. Example:

- Headlamp assembly with nested position lamp.

Main-light functions for Europe

Regulations and directives for Europe

The most important regulations and directives are laid down in ECE R112 [20], ECE R113 [21], ECE R48 [12], 76/756/EEC [24], ECE R98 [18], and ECE R123 [23].

- ECE R112: Headlamps for asymmetrical low beam and/or high beam that are fitted with filament bulbs or LED modules (cars, buses, trucks).
- ECE R113: Headlamps for symmetrical low beam and/or high beam that are fitted with filament bulbs or LED modules (motorbicycles, motorcycles).
- ECE R48 and 76/756/EEC: for attachment and application.
- ECE R98: Headlamps with gas-discharge lamps as per ECE R99.
- ECE R123: Adaptive front-lighting systems (AFS) for motor vehicles.

The installation regulations described in the following refer to passenger cars.

Figure 2: Headlamp systems

- Dual-headlamp system,
- Quad-headlamp system,
- Quad-headlamp system with additional fog lamps.

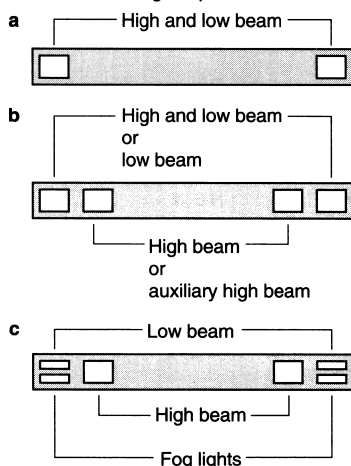
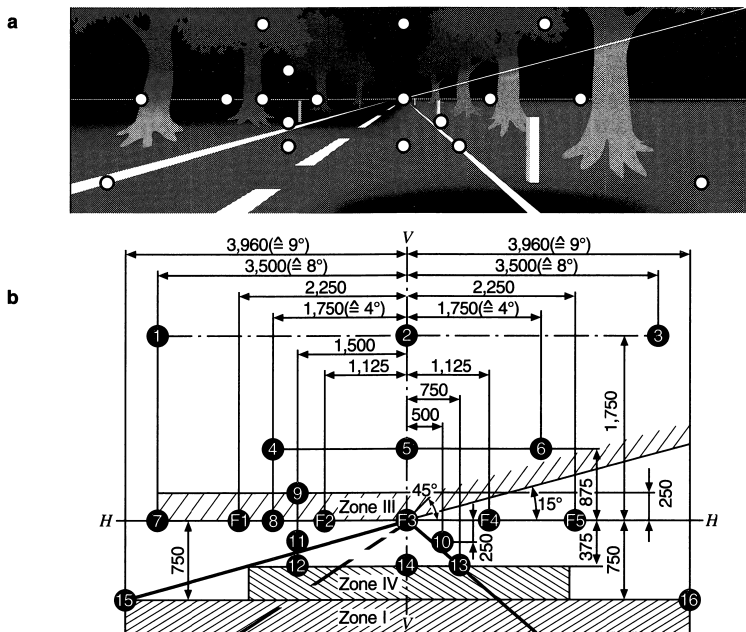


Figure 3: Luminous intensities of headlamps for Europe/ECE

- a) Road perspective from driver's viewpoint.
b) Measurement point relative to road perspective as per ECE R 112.



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Table 2: Measurement points and luminous intensities for headlamps

Low beam (dipped beam)				High beam				
Measurement points on graphic			Luminous intensity	Measurement points		Luminous intensity		
Figure no.	RHD traffic	LHD traffic	Class B (lx)		Figure no.	Point	Class B (lx)	
01	8L/4U	8R/H 4R/H B50R 75L 75R 50L 50R 50V 25L 25R	≤ 0.7		F1 F2 F3 F4 F5	$E_{\max}$	$48 < E$	
02	V/4U		≤ 0.7					< 240
03	8R/4U		≤ 0.7				$E_{H-5.15^{\circ}}$	> 6
04	4L/2U		≤ 0.7				$E_{H-2.55^{\circ}}$	> 24
05	V/2U		≤ 0.7				E_{HV}^9	≥ 0.8
06	4R/2U		≤ 0.7				$E_{\max}$	
07	8L/H		≥ 0.1; ≤ 0.7			$E_{H+2.55^{\circ}}$	> 24	
08	4L/H		≥ 0.2; ≤ 0.7			$E_{H+5.15^{\circ}}$	> 6	
09	B50L		≤ 0.4		For low beam: Total 1 + 2 + 3 ≥ 0.3 lx Total 4 + 5 + 6 ≥ 0.6 lx			
10	75R		≥ 12					
11	75L		≤ 12					
12	50L		≤ 15					
13	50R		≥ 12					
14	50V		≥ 6					
15	25L		≥ 2					
16	25R		≥ 2					
Any point in zone III				≤ 0.7		1) E is the current measured value in point 50R or 50L.		
Any point in zone IV				≥ 3				
Any point in zone I				≤ 2E ¹⁾				

Low beam, installation

Regulations prescribe 2 white-light low-beam headlamps for multiple-track vehicles (Figure 4).

Low beam, lighting technology

Automotive headlamp performance is subject to technical assessment and verification before they are put into volume production. Among the requirements are minimum luminous intensity, to ensure adequate road-surface visibility, and maximum intensity levels, to prevent glare (see measurement points and luminous intensities for headlamps, Figure 3 and Table 2).

Homologation testing is carried out under laboratory conditions using test lamps manufactured to more precise tolerances than those installed in production vehicles. The lamps are operated at the specified test luminous flux for each lamp category. The laboratory conditions apply across the board to all headlamps, but only take limited account of the specifics of individual vehicles, such as headlamp fitted height, vehicle power supply, and adjustment.

Low beam, switching

All high-beam headlamps must extinguish simultaneously when the low beams are switched on. Dimming (gradual deactivation) is permitted, with a maximum dimming period of 5 seconds. A 2-second response delay is required to prevent the dimming feature from activating when

the headlamp flashers are used. When the high-beam headlamps are switched on, the low-beam units may continue to operate (simultaneous operation). H4 bulbs are generally suitable for short periods of use with both filaments in operation.

High beam, installation

A minimum of two and a maximum of four headlamps are prescribed for the high-beam mode. The prescribed instrument-cluster high-beam indicator lamp is blue or yellow in color.

High beams, lighting technology

The high beam is usually generated by a light source located at the focal point of the reflector (Figure 5). This causes light to be reflected outward in the direction of the reflector axis. The maximum luminous intensity achievable by the high beam is largely a function of the reflector's illuminated area. In quad-headlamp systems, in particular, roughly paraboloid high-beam reflectors can be replaced by units with complex geometrical configurations designed to supply a "superimposed" high-beam pattern. The calculations employed to design these units seek to achieve a high-beam distribution that harmonizes with the low-beam pattern (simultaneous activation). The pure high beam is "superimposed" as it were on the low-beam projection. The annoying overlap area close to the front of the vehicle is done away with in this case.

Figure 4: European headlamp system (low beam)

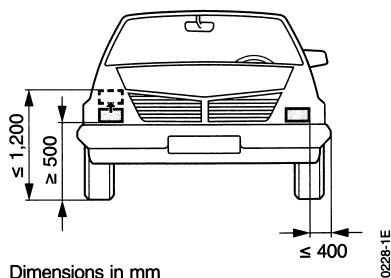
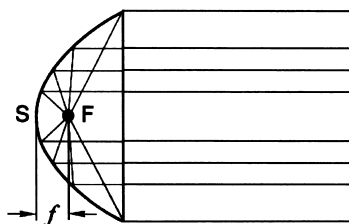


Figure 5: Parabolic reflector

F Focal point,
S Vertex of the parabola,
f Focal length.



High-beam light distribution pattern is defined in the regulations and guidelines together with stipulations governing the low beams.

The maximum approved luminous intensity, a composite of the intensity ratings of all high-beam headlamps installed on the vehicle, is 430,000 cd. This value is indicated by identification codes located adjacent to the homologation code on each headlamp. 430,000 cd corresponds to the figure 100. The luminous intensity of the high beam is indicated by the number, e.g. 25, stamped next to the round ECE approval mark. If these are the only headlamps on the vehicle (no auxiliary driving lamps), then the composite luminous intensity is in the range of 50/100 of 430,000 cd, i.e. 215,000 cd.

Auxiliary driving lamps

Auxiliary driving lamps are used to complement the effectiveness of the high beam in standard high-beam headlamps.

Auxiliary driving lamps are mounted and aimed in the same way as standard headlamps, and the underlying lighting technology is the same. Auxiliary driving lamps are also subject to the regulations governing maximum luminous intensity in vehicle lighting systems, according to which the sum of the reference numbers of all headlamps fitted to a vehicle must not exceed 100. For older headlamps without approval number, the number 10 is used for general assessment purposes.

Main-light functions for the USA

Regulations and directives

The national standard is the Federal Motor Vehicle Safety Standard (FMVSS) No.108 [27] and the SAE Ground Vehicle Lighting Standards Manual (Standards and Recommended Practices) to which it refers.

The regulations governing installation and control circuits for headlamps are comparable in parts to those in Europe. Since 5/1/1997, however, headlamps with light-dark boundaries have also been authorized in the USA, but these require manual adjustment. It is now possible to develop headlamps which conform to legal requirements in both Europe and the USA.

As in Europe, dual- and quad-headlamp systems are used in the USA. The fitting and use of fog lamps and additional high-beam headlamps, however, are subject to a variety of, in some cases, widely diverging, local laws passed by the 50 individual states.

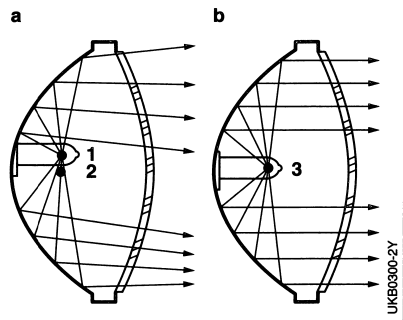
Up to 1983 the allowable sealed-beam headlamp sizes (Figure 6) in the USA were as follows:

Dual-headlamp systems:

- 178 mm diameter (round),
- 200 × 142 mm (rectangular).

Figure 6: American sealed-beam headlamps

- a) Low beam,
- b) High beam.
- 1 Low-beam filament,
- 2 Focal point,
- 3 High-beam filament (at focal point).



Quad-headlamp systems:

- 146 mm diameter (round),
- 165 × 100 mm (rectangular).

Low beam, lighting technology

The light-pattern requirements in America differ to a greater or lesser degree from the European system depending on the design type. In particular, the minimum glare levels are higher in the USA, and the maximum low-beam width is closer

to the vehicle. The basic setting is generally higher (see measurement points in Figure 7 and Table 3).

High beam

The designs for high-beam headlamps are the same as in Europe. Differences exist in the required dispersion width of the light pattern, and there is a lower maximum figure on the axis of the high-beam headlamp.

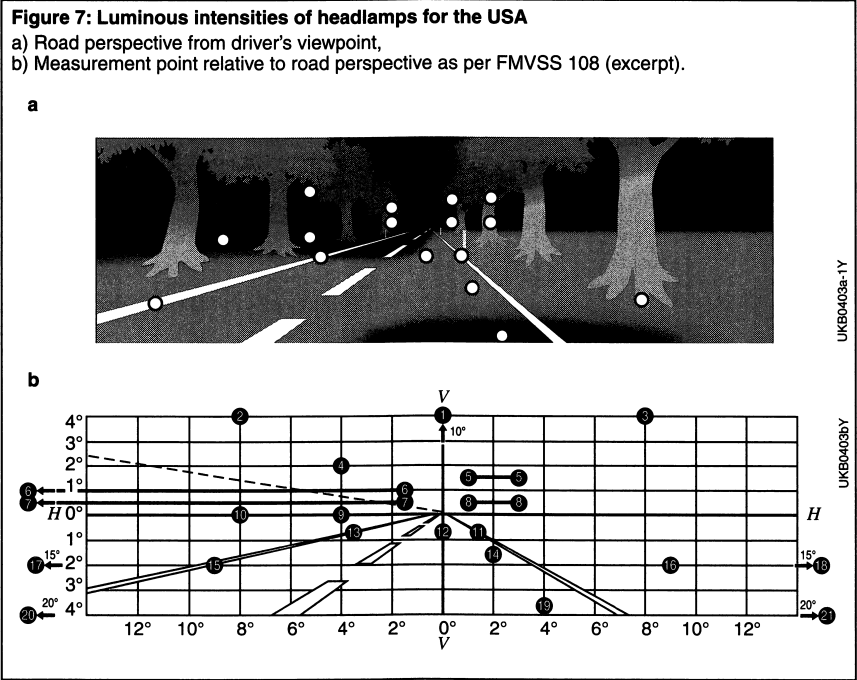


Table 3: Measurement points and luminous intensities for headlamps, low beam

Figure no.	Measurement points	Luminous intensity (cd)	Figure no.	Measurement points	Luminous intensity (cd)
01	10U-90U	≤ 125	11	0.6D, 1.3R	≥ 10,000
02	4U, 8L	≥ 64	12	0.86D, V	≥ 4,500
03	4U, 8R	≥ 64	13	0.86D, 3.5L	≥ 1,800; ≤ 12,000
04	2U, 4L	≥ 135	14	1.5D, 2R	≥ 15,000
05	1.5U, 1R-3R	≥ 200	15	2D, 9L	≥ 1,250
05	1.5U, 1R-R	≤ 1,400	16	2D, 9R	≥ 1,250
06	1U, 1.5L-L	≤ 700	17	2D, 15L	≥ 1,000
07	0.5U, 1.5L-L	≤ 1,000	18	2D, 15R	≥ 1,000
08	0.5U, 1R-3R	≥ 500; ≤ 2,700	19	4D, 4R	≥ 12,500
09	H, 4L	≥ 135	20	4D, 20L	≥ 300
10	H, 8L	≥ 64	21	4D, 20R	≥ 300

Designs

Sealed-beam design

In this design, which is no longer used (Figure 6), the aluminized glass reflector and the lens must be sealed gas-tight on account of the light sources that are not encapsulated. The whole unit is sealed and filled with an inert gas. If a filament burns through, the entire light source must be replaced. Units with halogen lamps are also available.

The limited range of available sealed-beam headlamps on the market severely restricted the freedom of design for vehicle front ends.

Vehicle headlamp aiming device (VHAD)

This design involves replaceable-bulb headlamps which are adjusted vertically with the aid of a spirit levels integrated in each headlamp, and horizontally by means of a system comprising a needle and dial. In fact, this is equivalent to "on-board aiming".

Headlamps for visual aim (VOL / VOR)

These systems have been in use since 1997. These are replaceable-bulb headlamps whose low beam has a light-dark boundary line which allows visual aiming of the headlamps (as is standard in Europe).

Either the left horizontal light-dark boundary (VOL, visual optical aim left; VOL mark on the headlamp) or, as is more often the case in the USA, the right horizontal light-dark boundary (VOR, visual optical aim right; VOR mark on the headlamp) is used. What is notable about the US systems is the position of the light-dark boundary, which is much closer to the horizon (inclination depending on type 0.4 to 0%). This increases the potential dazzling risk of such systems.

There is no horizontal aiming with this type of headlamp.

Definitions and terms

Photometrical terms and definitions

Headlamp range

This is the distance at which the light beam continues to supply a specified luminous intensity – mostly the 1 lux line at the right side of the road (LHD traffic).

Geometric range of a headlamp

This is the distance to the horizontal portion of the light-dark boundary on the road surface (see Table 4). A low-beam inclination of 1%, or 10 cm/10 m, results in a geometric range equal to 100 times the headlamp's fitted height (as measured between the center of the reflector and the road surface).

Visual range

The visual range is the distance at which an object (vehicle, object, etc.) within the luminance distribution of the human visual field is still visible.

The visual range is influenced by the shape, size, and reflectance of objects, the road-surface type, headlamp design and cleanliness, and the physiological condition of the driver's eyes. Due to the large number of influencing factors, it is not possible to quantify this range using precise numerical definitions. Under extremely unfavorable conditions (with RHD traffic, on the left side of a wet road surface) the visual range can fall to below 20 m. Under optimum conditions, it can extend outward to more than 100 m (with RHD traffic, on the right side of the road).

Disability glare

This is the quantifiable reduction in visual performance that occurs in response to light sources emitting glare. An example would be the reduction in visual range that occurs as two vehicles approach one another.

Discomfort glare

This condition occurs when a glare source induces discomfort without, however, causing an actual reduction in visual performance. Discomfort glare is assessed according to a scale defining different levels of comfort and discomfort.

Headlamp technology

Reflector focal length

Conventional reflectors for headlamps and other automotive lamps are usually parabolic in shape (Figure 5). The focal length f (distance between the vertex of the parabola and the focal point) is 15 to 40 mm.

Free-form reflectors

The geometrical configurations of free-form reflectors are generated using complex mathematical calculations (HNS, Homogeneous Numerically Calculated Surface). Here, the low mean focal length f is defined relative to the distance between the reflector vertex and the center of the filament. Typical values range from 15 to 25 mm.

In the case of reflectors partitioned with steps or facets, each partition can be created with its own mean focal length f .

Reflector illuminated area

This is the parallel projection of the entire reflector opening on a transverse plane. The standard reference plane is perpendicular to the vehicle's direction of travel.

Effective luminous flux, efficiency of a headlamp

The first of the above is the portion of the light source's luminous flux that is capable of supplying effective illumination via its reflective or refractive components (for instance, as projected on the road surface via the headlamp reflector). A reflector with a short focal length makes efficient use of the filament bulb and has high efficiency because the reflector extends outward to encompass the bulb, allowing it to convert a large proportion of the luminous flux into a useful light beam.

Angles of geometric visibility

These are angles that are defined relative to the axis of the lighting device, at which the illuminated area must be visible.

Technical design variations of headlamps

Components

Reflector

Reflectors direct the light from the light source either directly onto the road (reflection system) or into an intermediate plane which is further projected by a lens (projection system). The reflectors are made of plastic, die-cast metal or sheet steel.

Plastic reflectors are manufactured by injection molding (thermosetting plastics), which offers a considerably better precision of geometry reproduction than the deep-drawing process. The geometric tolerances that can be achieved lie in the range of 0.01 mm. Furthermore, reflectors in stages and any desired facet distributions can also be implemented. The base material requires no corrosion-proofing treatment.

The die-cast metal used is usually aluminum, or occasionally magnesium. The advantages are high thermal resistance and the ability to produce shapes with a high degree of complexity (shaped bulb holders, screw holes and bosses).

The surfaces of thermosetting plastic and die-cast metal reflectors are given a smooth finish by spray painting or powder-based paint before a layer of aluminum 50 to 150 nm thick is applied. An even thinner transparent protective coat prevents the aluminum from oxidizing.

Sheet-steel reflectors are manufactured using deep-drawing and punching dies. A powder-based paint is then applied. This process hermetically seals the sheet steel and gives it excellent surface smoothness. The base coat created in this way is, like other reflectors, coated with aluminum.

Lens

A large proportion of contoured lenses is manufactured using high-purity glass (free of bubbles and streaks). During the lens molding process, high priority is given to surface quality in order to prevent undesirable upward light deflection in the final product – this would tend to create glare for oncoming traffic. The type and configuration of the lens prisms depend

on the reflector and the desired light distribution pattern.

The clear lenses used on modern headlamps are usually made of plastic. Besides reducing weight, plastic lenses provide other advantages for automotive applications, including greater freedom in headlamp and vehicle design. Since around 2007 multicolor plastic lenses (2-component lenses) have also been used in which the edge area is sprayed in a different color, usually black or gray. The advantage here is that the spray tools are designed in such a way that no inner slides are needed and therefore no separating lines are created on the visible surface. Light dispersion from the edge areas is also avoided.

There are several reasons why plastic lenses should not be cleaned with a dry cloth:

- Despite the scratch-proof coating, rubbing with a dry cloth can damage the surface of the lens.
- Rubbing with a dry cloth can produce an electrostatic charge in the lens, which can then allow dust to build up on the inside of the lens.

Headlamp versions

Reflection headlamps

For conventional headlamp systems with virtually parabolic reflectors (Figure 5 and Figure 8), the quality of the low beam increases in direct proportion to the size of the reflector. At the same time the geometric range increases as a function of installation height.

These factors must be balanced against the aerodynamic constraints according to which the vehicle's front-end profile must be kept as low as possible. Under these circumstances, increasing the size of the reflector results in wider headlamps.

Reflectors of a given size, but with different focal lengths, also perform differently. Shorter focal lengths are more efficient and produce wider light beams with better close-range and side illumination. This is of particular advantage during cornering.

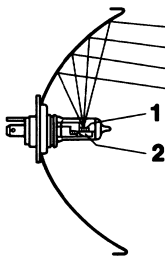
Specially developed lighting programs (CAL, computer-aided lighting) enable the implementation of infinitely variable reflector shapes with non-parabolic sections and of faceted reflectors.

Headlamps with facet-type reflectors

In the case of facets, the reflector surface is partitioned and each individual segment is individually optimized. The im-

Figure 8: Low beam (beam-path vertical section, H4 lamp)

- 1 Low-beam filament,
- 2 Bulb cover.

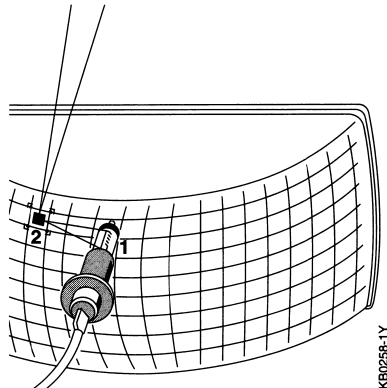


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Figure 9: Free-form or facet-type reflector

Filament pattern reflection with mirror optics.

- 1 Filament,
- 2 Mirror optics.



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portant feature of surfaces with facet-type reflectors is that discontinuity and steps are permitted at all boundary surfaces of the partition. This results in freely shaped reflector surfaces with maximum homogeneity and side illumination (Figure 9 and Figure 10).

PES headlamps

The PES headlamp system (poly-ellipsoid system) employs imaging optics (Figure 11) and offers greater design scope than conventional headlamps. A lens aperture area with a diameter of only 40 to 70 mm allows the generation of light patterns pre-

viously only achievable with large-area headlamps. This result is obtained using an elliptical (CAL designed) reflector in combination with optical projection technology. A screen reflected with the objective projects precisely defined light-dark boundaries. Depending on specific individual requirements, these transitions can be defined as sudden or gradual intensity shifts, making it possible to obtain any geometry required.

PES headlamps can be combined with conventional high beams, position lamps and PES fog lamps to form lighting-strip units in which the entire headlamp is no higher than approximately 80 mm.

In PES headlamps, the beam path can be configured in such a way that the surroundings of the objective are also used in the signal image. This enlargement of the signal image is used above all with small objective diameters to reduce the psychological glare for the oncoming traffic. The additional surface can also be made into a lens, a partially vapor-deposited screen or a design feature with illuminated round or rectangular gaps, or illuminated three-dimensional objects.

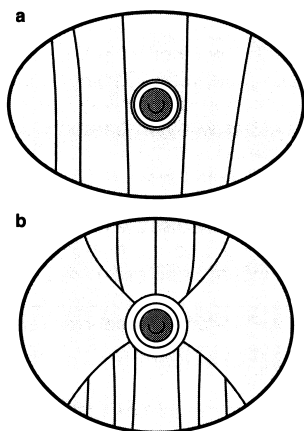
Xenon headlamps

The headlamp system with a xenon gas-discharge lamp as its central component (Figure 12) generates high illumination-intensity levels with minimal frontal-area requirements, making it ideal for vehicles with aerodynamic styling with exceptional c_w values. In contrast with the conventional filament bulb, light is generated by a plasma discharge inside a burner the size of a cherry stone (Figure 13).

The arc of the 35W xenon bulb D2S delivers, when compared with the H7 bulb, twice the luminous flux with a higher color temperature (4200 K), i.e., the color of the light is very similar to natural sunlight. A D5S bulb with a power rating of only 25 W still delivers three times the luminous flux of an H7 bulb with the same high color temperature as a 35W xenon bulb. Maximum luminous efficiency, corresponding to approx. 90 lm/W, is available as soon as the quartz element reaches its operating temperature of more than 900 °C. Brief high-power operation at currents of up to 2.6 A (continuous operation: approx.

Figure 10: Facet-type reflector

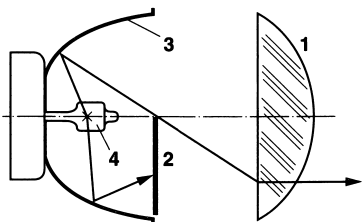
- a) Vertically partitioned,
- b) Radially and vertically partitioned.



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Figure 11: PES reflector (optical principle)

- 1 Objective,
- 2 Screen,
- 3 Reflector,
- 4 Lamp.



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0.4 A) can be used to obtain "instant light". 2000 hours of service life are sufficient for the average required total duration of operation in passenger cars. As no sudden failure occurs as in the case of a filament, diagnosis and replacement in good time are possible.

At present gas-discharge lamps with the type designations D1 and D2 and since 2012 only the type designations D3 and D4 are used in the power-rating

range of 35 W. In the D3 and D4 families dosing of the heavy metal mercury (approximately 1 mg) can be dispensed with. These lamps are distinguished by lower lamp voltage, a different plasma composition, and different arc geometries. The same applies to the new categories of 25W gas-discharge lamps D5, D6 and D8. The electronic control units for the individual lamp types are generally developed for a specific design type and are not universally interchangeable.

The D2 and D4-series automotive gas-discharge lamps feature high-voltage-proof bases and UV glass shielding elements. On the D1 and D3-series models, the high-voltage electronics necessary for operation are also integrated in the lamp base. Category D5 lamps are a special case – these lamps have the advantage that both the ignition unit and the control unit are already integrated in the base of the gas-discharge lamp.

All systems feature two subcategories: S-lamps for projection-system headlamps and R-lamps for reflection headlamps with an integrated light shield for producing the light/dark boundary comparable with the bulb cover used for halogen H4 low beams. At present the D1S and D3S lamps are the type most widely used. An integral part of the Litronic headlamp (contraction of Light and Electronics) is the electronic ballast unit responsible for activating and monitoring the lamp (Figure 14). Its functions

Figure 12: Gas-discharge lamp D5S

- 1 UV glass shielding bulb,
- 2 Lead-in insulator,
- 3 Discharge chamber (burner),
- 4 Electrodes,
- 5 Bulb base,
- 6 Integrated ignition and control unit.

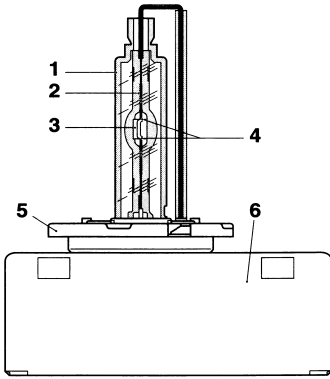
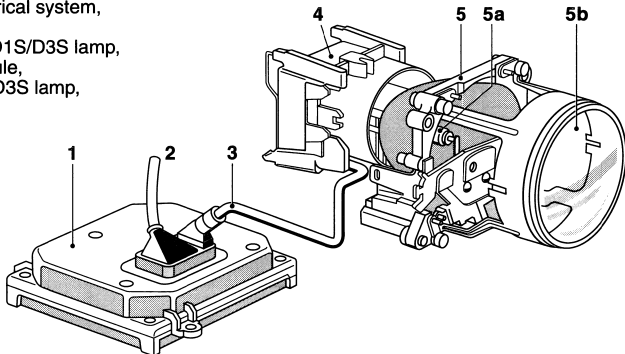


Figure 13: System components for PES-design Litronic headlamp

- 1 ECU,
- 2 To vehicle electrical system,
- 3 Shielded cable,
- 4 Ignition unit of D1S/D3S lamp,
- 5 Projection module,
- 5a Burner of D1S/D3S lamp,
- 5b Lens.



include ignition of the gas discharge (voltage 10 to 20 kV), controlled power supply during the warm-up phase when the lamp is cold, and demand-oriented supply during stationary operation.

The system furnishes largely consistent levels of illumination (i.e., elimination of luminous-flux changes) by compensating for fluctuations in the vehicle system voltage. If the lamp goes out (for instance, due to a momentary voltage drop in the vehicle electrical system), re-ignition is spontaneous and automatic.

The electronic ballast unit responds to defects (such as a damaged lamp) by interrupting the power supply to help avoid injury in the event of contact.

The xenon light emitted by Litronic headlamps produces a broad carpet of light in front of the vehicle combined with a long range (Figure 15). This has made it possible to achieve substantially wider road-illumination patterns for lighting the edges of bends and wide roads as effectively as a halogen unit illuminates straight stretches of road. The driver enjoys substantial improvements in both visibility and orientation in difficult driving conditions and bad weather.

In accordance with ECE Regulation 48 [12], Litronic headlamps with 35W gas-discharge lamps are combined with automatic headlamp leveling control and headlamp washer systems. These requirements do not apply to headlamps

with 25W lamps on account of the lower luminous flux of the lamps, i.e., such headlamps may also – like halogen headlamps – be operated with manual headlamp leveling control. This makes it

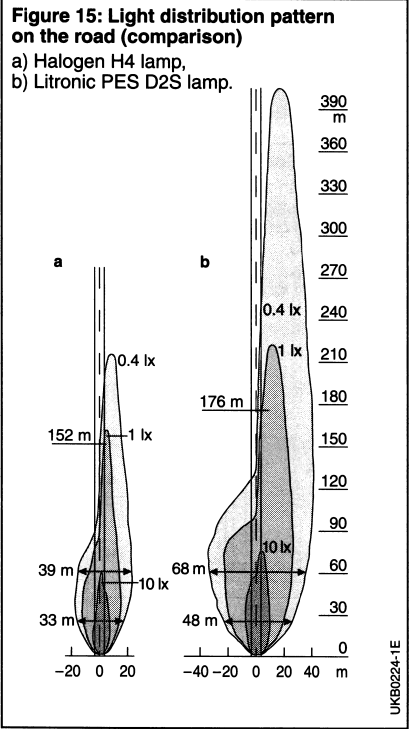
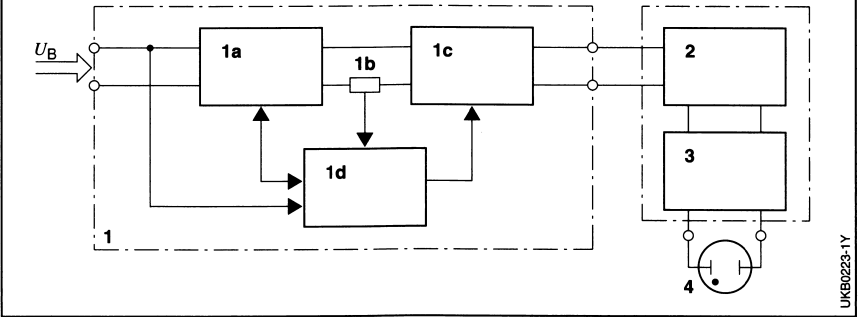


Figure 14: Electronic ballast unit for 400-Hz AC supply and pulse triggering of the bulb

1 ECU, 1a DC/DC converter, 1b Shunt, 1c DC/AC converter, 1d Microprocessor, 2 Ignition unit, 3 Lamp socket, 4 D2S lamp, U_B Battery voltage.



easier to switch from halogen to 25W gas-discharge headlamps in the aftermarket sector.

Bi-Litronic

Bi-Litronic systems permit both the low and high beams to be generated by the arc of a single gas-discharge lamp.

Bi-Litronic "Projection" (bi-xenon)

The Bi-Litronic "Projection" system is based on a PES Litronic headlamp (Figure 16). For the high beam the light shield for creating the light/dark boundary (for the low beam) is moved out of the light beam. With lens diameters of 70 mm, Bi-Litronic "Projection" allows the present highly compact headlamp design with combined high and low beam and at the same time outstanding luminous efficiency (Figure 18).

The primary advantage of Bi-Litronic "Projection" is the xenon light for high-beam operation.

Bi-Litronic "Reflection"

Both mono- and bi-xenon systems use a single DxR lamp for both headlamp functions.

In the bi-xenon version, when the high-/low-beam switch is operated, an electro-mechanical actuator moves the gas-discharge lamp in the reflector into the appropriate position for producing the high or low beam as required (Figure 17).

Figure 16: Bi-Litronic "Projection"

1 Low beam,
2 High beam.

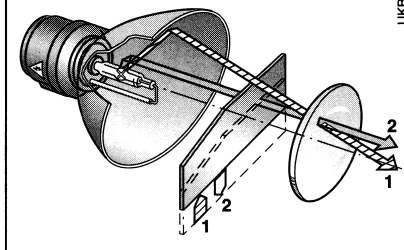


Figure 17: Bi-Litronic "Reflection"

1 Low beam,
2 High beam.

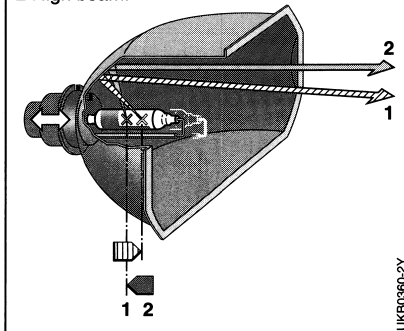
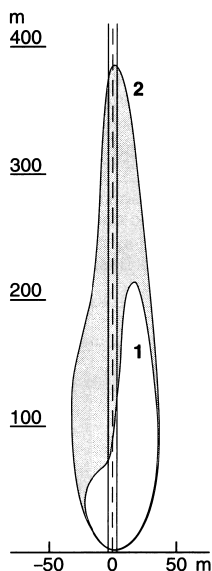


Figure 18: Light distribution patterns of Bi-Litronic

1 Low beam,
2 High beam.



Cornering headlamps

Cornering headlamps have been approved for public use since the beginning of 2003. Whereas previously only high-beam headlamps were allowed to turn in response to changes in steering angle (1960s Citroën DS), swiveling low-beam headlamps are now also permitted (dynamic cornering headlamps or adaptive headlamps) or a supplementary light source (static cornering headlamps). This provides for a larger visual range on winding roads.

Static cornering headlamps

Static cornering headlamps are used mainly to illuminate areas close to the side of the vehicle (switchbacks, turning maneuvers). For this purpose, activating additional reflector elements is generally the most effective way.

Dynamic cornering headlamps

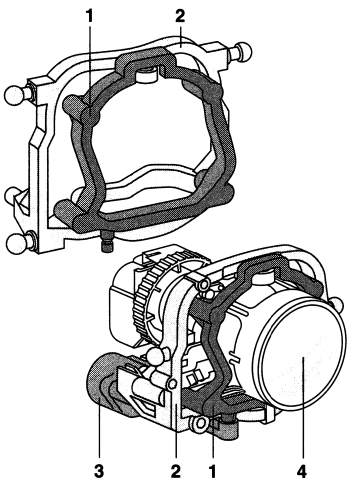
Dynamic cornering headlamps are used to illuminate the changing course of the road, e.g., on winding overland highways (Figure 19).

In contrast with the directly linked swiveling action of the cornering headlamps of the 1960s, modern "high-end systems" electronically control the rate of swivel and the swivel angle in response to the vehicle's speed. This optimizes "harmonization" between the headlamps and the vehicle attitude, and eliminates "jerky" headlamp movements. Headlamp positioning is performed by a positioner unit

(stepping motor) which moves the basic or low-beam module or the reflector elements in response to changes in steering-wheel angle or the steering angle of the front wheels (Figure 20). Sensors detect those movements to prevent glare for oncoming traffic by means of "failsafe" algorithms. General legal requirements specify that the headlamp beam may only

Figure 20: Cornering-headlamp modules

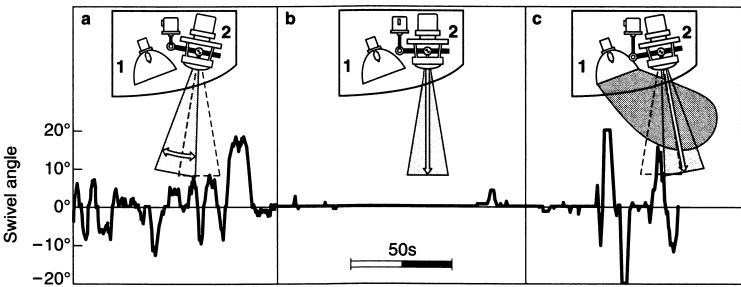
- 1 Supporting frame,
- 2 Mounting frame,
- 3 Drive motor for horizontal rotation,
- 4 Bi-Litronic PES.



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Figure 19: Switch and aiming strategy of turning and basic modules of a static/dynamic cornering headlamp (left side)

a) "Highway/cornering" position, b) "Expressway" position, c) "City/turning" position.
1 Turning module, 2 Basic module.



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be turned as far as the center line of the road at a distance of approx. 70 m in front of the vehicle to prevent glare for oncoming vehicles.

Road safety and driving convenience

The introduction of dynamic cornering headlamps is a significant improvement in the safety and convenience of driving at night (Figure 21). Compared with conventional low-beam headlamps, improvements in visual range of approximately 70% are achieved, representing an extra 1.6 seconds of travel time. With cornering headlamps, a motorist can assess hazards better and start braking sooner. As a result, the severity of an accident can be significantly reduced. Static cornering headlamps double the visual range for turning maneuvers.

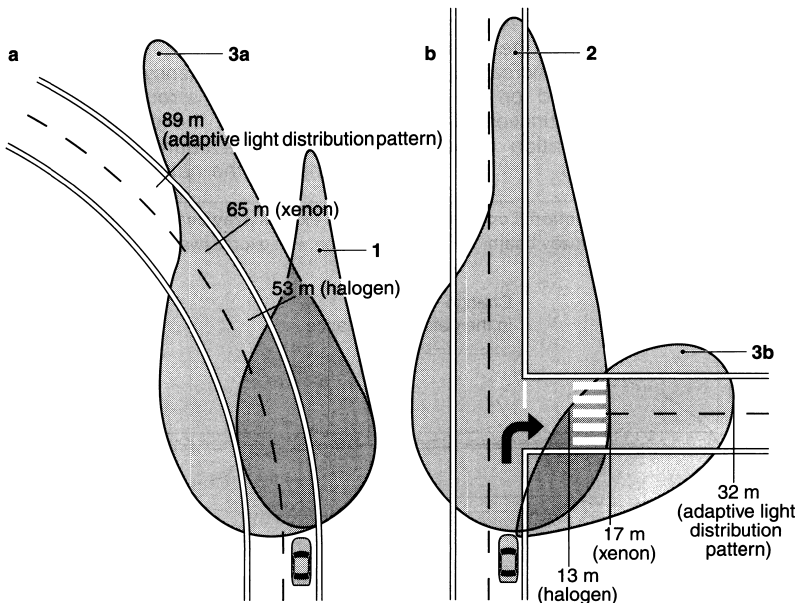
AFS functions

Expressway beam

For special driving situations, altered light patterns (AFS, adaptive front-lighting system) have been developed to enable a better view for the driver for each driving state. In the development of the expressway beam (Figure 22), special attention was paid to achieving a better range for the driver without dazzling oncoming traffic to an excessive degree. The increase in the detection distance to up to 150 m enables an extension of driving time up to the detected object of approx. 2 seconds (comparison at 100 km/h with halogen headlamps). This enables the driver to better assess a critical situation and, possibly, to initiate braking much earlier.

Figure 21: Measurable improvement of visibility for driver with adaptive light distribution pattern of cornering headlamps

- a) Left-hand bend, dynamic cornering headlamps,
- b) Right turn, static cornering headlamps.
- 1 Halogen headlamps, 2 Xenon headlamps,
- 3a Adaptive light distribution pattern with dynamic cornering headlamps,
- 3b Adaptive light distribution pattern with static cornering headlamps.



Poor-weather light

In the case of the poor-weather light beam, the special focus is on improving the optical guidance on the road. Especially the zones around the sides of the road are better illuminated.

Most variants of the poor-weather light feature movement of the left cornering-headlamp module by 8° to the side and simultaneously a slight lowering or activation of the static cornering lights. This provides for very wide illumination of the road and the edge of the road. In future, component elements, for example the elements responsible for widened side illumination, will be activated sequentially. Control parameters are, for example, steering-angle information and direction-indicator actuation. The individual segments are then "quasidynamically" activated.

Light functions and driver-assistance systems

The introduction of video technology in automotive engineering makes it possible to implement camera-based headlamp functions as well. When the position of an oncoming vehicle is identified by the camera, the headlamp or the AFS system can adapt the range of the driving lights in such a way that it is increased for large distances and reduced for smaller distances (dynamic range function). This ensures optimum illumination without dazzling oncoming traffic.

LED headlamps

Potential for reducing energy consumption

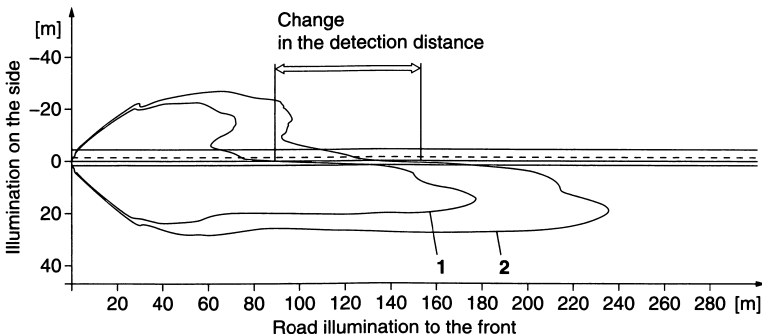
LEDs are increasingly being used as economical alternatives in avoiding CO₂ emissions and fuel consumption. The energy consumption of the main-light functions will play a significant role in future low-energy vehicles. Xenon and LED alternatives offer the levels of optimized consumption and improved road safety called for by the EU.

Today's LED systems already consume, depending on performance (luminous flux, range, side illumination), much less energy than halogen bulbs. Power consumption currently fluctuates between 28 W and 50 W per headlamp. When compared with bulb power output of approx. 65 W (for 13.2 V), this means potential savings of 30 to 70 W per vehicle.

One projection system and two reflection elements are used for the low-beam spot. The light of three multichip LEDs each with two LEDs is concentrated by three primary optical elements and imaged by a projection lens. The optical system incorporates a screen to guarantee the quality of the light-dark boundary. A reflector is positioned both above and below the lens (Figure 23).

The optical efficiency of the LED low-beam headlamp is roughly 45%. Compared with this, the efficiency of a bi-xenon system is around 33%. This can be explained by the nature of LEDs, which

Figure 22: Metrological-functional connection between expressway beam and highway beam
1 Road illumination with highway beam, 2 Road illumination with expressway beam.



give off their light only to the half-space and do not, like conventional light sources, illuminate the entire space. Because of the higher efficiency of an LED system, less luminous flux is needed in the LEDs to deliver the same luminous flux to the road with one LED low-beam headlamp.

The better LEDs will become in future, the less power will be output by the control unit, the light performance always being kept at a constantly high level.

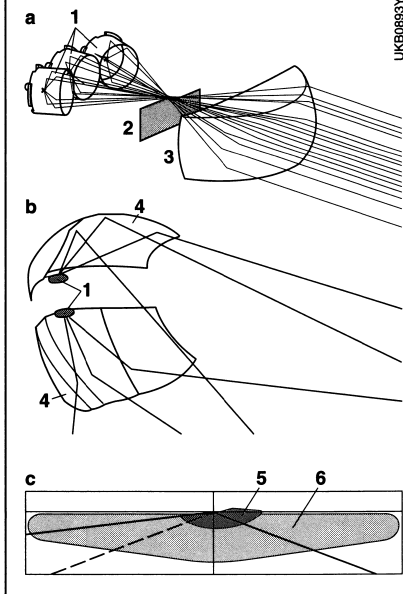
The first LED headlamps on the market also show the increasing role of design elements in headlamps. In 2008, for example, a headlamp received an internationally renowned design award for the first time.

Figure 23: Basic design of an LED low-beam headlamp

Interaction of projection and reflection systems.

- a) Projection system for the spot,
- b) Reflection system for the basic light,
- c) Total light distribution.

- 1 LEDs,
- 2 Light shield for light-dark boundary,
- 3 Projection system,
- 4 Reflector,
- 5 Spot,
- 6 Basic light.



Around 75% of total worldwide mileage is completed during daytime hours. Great importance is therefore also attached to the energy consumption of the daytime driving lights function. A typical LED daytime driving light has an energy consumption of 14 W (0.36 g CO₂/km) per vehicle. Use of vehicle lights during the day consumes up to 300 W (7.86 g CO₂/km) (low-beam headlamps, tail lamps, position lamps, number-plate lamps, switch and instrument lighting).

Laser light

Functions in the headlamp whose light source is based on a blue laser diode (see Laser LED) are called laser light. Similarly to white LEDs the blue laser radiation is converted with a phosphor conversion into a white wavelength spectrum and diffused. The advantage of the laser-light source including converter is the five to seven times higher luminance compared with the LEDs currently used. The higher luminance is used to generate light distributions with high luminous intensities with smaller reflectors. This can be used for example to produce an auxiliary high beam which roughly doubles the range on the road. The higher luminous intensities on the road lead to improved, earlier detection of objects at great distances and represent a significant increase in safety.

Glare-free high beam

“Glare-free high beam” is a light function which is classed as part of the high beam but in contrast to the conventional high beam can also be activated in traffic situations with preceding or oncoming traffic. This function is referred to in the international context and in the statutory provisions as Adaptive Driving Beam (ADB). Activation requires a camera and image analysis which determines the positions of the other road users quickly and precisely. Shadows in the high-beam distribution are specifically controlled with this information (Figure 24, see also High-beam assistant).

Dynamic system

This type is based on the dynamic cornering headlamps (see Figure 20). Here the left headlamp generates a partial high-beam distribution with a vertical light/dark boundary to the right side and the right headlamp generates a right partial high-beam distribution. In the “normal” high beam both the partial high-beam distributions are superimposed in the middle to achieve high luminous intensities in the central range. To prevent other road users from being dazzled, a shadow is generated in the corresponding position and width with the dynamic movement of both sides.

Figure 24: Glare-free high beam (1st generation)

Schematic diagram.
V Vertical,
H Horizontal.

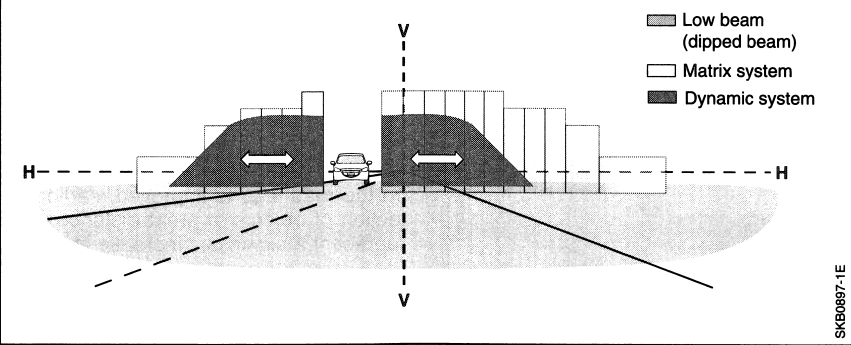
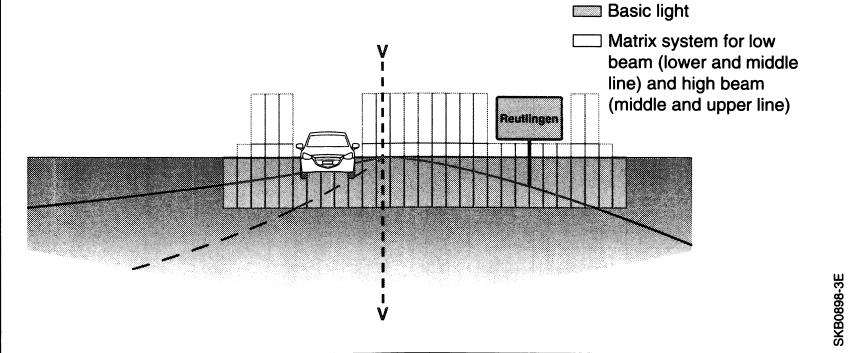


Figure 25: Matrix headlamp (2nd generation)

Schematic diagram.



Matrix system

In realizing the glare-free high beam as a matrix system the light distribution is made up typically of 12 to 18 strips. Individual LEDs or LED arrays with individually switchable chips are used as light sources and the specific attributes such as compact shape, rapid switch-on and switch-off, and dimmability are used. Each strip is formed by an LED chip. The shadow adapted to the traffic situation is formed by the specific switching-off of individual LEDs or groups of LEDs. For "normal" high-beam distribution all the LEDs are operated and a balanced light distribution is created by means of a dimming profile.

Matrix headlamps

Three-row modules with 84 pixels are available in some vehicles as a further development of the one-row matrix systems for glare-free high beam. The shadows for fading out other road users can be positioned even more precisely with the finer pixel resolution (Figure 25). Further shadows can additionally be created for example to fade out road signs.

Matrix in the low beam

The essential difference from the one-row systems is that with this type part of the low beam is also formed with the matrix module. This enables even the dynamic cornering headlamps - without mechanical actuators - to be implemented by purely electronic means via activation of the LEDs. In addition, the specific dimmability of areas of the light distribution can be used in adverse weather light to reduce the dazzling of oncoming traffic via the reflection on the wet road.

Fog lamps

Fog lamps (white light) are intended to improve orientation in fog, snow, heavy rain, and dust. A beam of light with a particularly high side dispersion is generated for this purpose. This ensures that the side of the road which is close to the vehicle is particularly well illuminated. The brightness levels achieved on close objects are significantly higher. Unlike the usually dark road surface far ahead of the vehicle, these high brightness levels help drivers to find their bearings better despite the poor weather conditions.

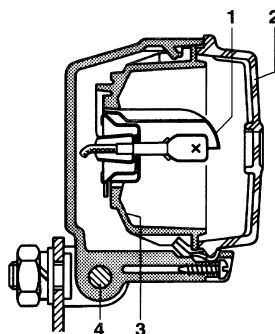
Designs

DIY and dealer-installed fog lamps are designed as individual projection units in their own housings. These are installed either upright on the fender, or suspended from it (Figure 26). Stylistic and aerodynamic considerations have led to increased use of integrated fog lamps, designed either for installation within body openings or included as a component within a larger light assembly (with adjustable reflectors when the fog lamps are combined with the main headlamps).

Present fog lamps produce white light. There is no substantive evidence that yellow lamps provide any physiological benefits. A fog lamp's effectiveness depends on the size of the illuminated area and the focal length of the reflector. Assuming the same illuminated area and focal length, the differences between round and

Figure 26: Fog lamp (upright mounting)

- 1 Bulb cover, 2 Lens, 3 Reflector, 4 Vertical adjustment spindle.



rectangular fog lamps from the technical viewpoint are negligible.

Regulations

Design is governed by ECE R19 [8], installation by ECE R48 [12] (and StVZO (Road Traffic Licensing Regulations) Article 52 [3] in Germany); two white or yellow fog lamps are permitted. The control circuit for switching the fog lamps must be independent of the high- and low-beam circuits. In Germany, the StVZO allows fog-lamp installation in positions more than 400 mm from the widest point of the vehicle's width, provided they are wired so that they can only be switched on when the low-beam headlamps are on (Figure 27).

Paraboloid with lens

A parabolic reflector featuring a light source located at the focal point reflects light along a parallel axis (as with the high-beam headlamp), and the lens extends this beam to form a horizontal band (Figure 26). A special bulb cover prevents the beam from being projected upward.

Free-form technology

Calculation methods, such as CAL (computer-aided lighting), can be used to design reflector shapes in such a way that they scatter light directly (i.e. without optical lens contouring) and also gener-

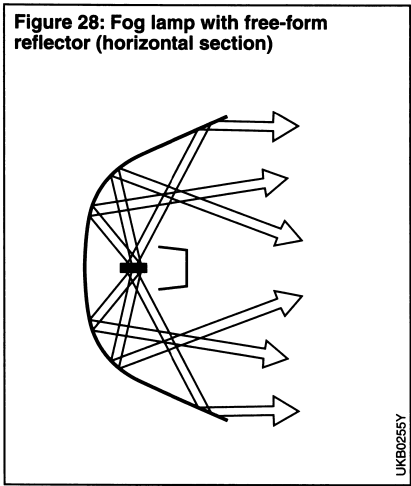
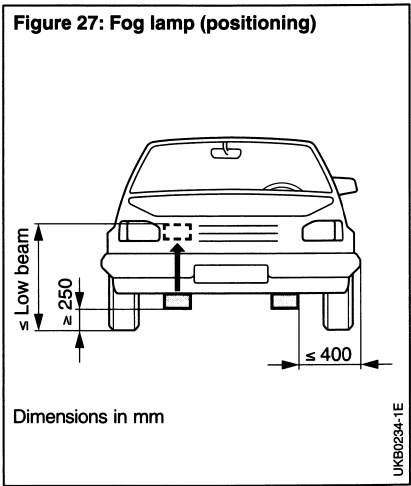
ate (without separate shading) a sharp light-dark boundary. The fact that the lamp features pronounced envelopment of the bulb leads to an extremely high volume of light combined with maximum dispersion width (Figure 28).

PES fog lamps

This technology minimizes reflected glare in fog. The screen, the image of which is projected onto the road surface by the lens, provides a light-dark boundary with minimum upward light dispersion.

Innovations

The technical-functional importance of the fog lamp has been pushed into the background slightly by the introduction of powerful halogen, xenon and AFS systems. The emphasis is still on the demand for improved lighting in traffic situations with adverse ambient conditions. This can be solved by separate fog lamps of the type used up to now (as per ECE R19 [8]), by the poor-weather light function of AFS systems (as per ECE R123 [23]), or by combinations of the two functions.



Installation and regulations for signal lamps

These types of lamp are intended to facilitate recognition of the vehicle, and to alert other road users to any intended or present changes in direction or motion. Uniform, distinctive colors in the red, yellow or white color range are prescribed for these lamps to denote their application. White, yellow and red lights are used to mark the positions of the vehicle at the front, the sides and the rear respectively. Stop lamps and rear fog warning lamps are also red. Yellow lights are used in most applications for turn-signal lamps. Only in the USA are red turn-signal lamps also permissible at the rear.

As projected along the reference axis, minimum and maximum luminous intensities for all lamps must remain within a range calculated to guarantee signal recognition without, however, causing glare nuisance for other road users.

Turn-signal lamps and hazard-warning flashers

(as per ECE R6 [6])

ECE R48 and 76/756/EEC specify Group 1 (front), Group 2 (rear), and Group 5 (side) turn-signal lamps for vehicles with three or more wheels. For motorcycles, Group 2 turn-signal lamps are sufficient.

The lamps are electrically monitored. A function indicator is required inside the vehicle. The dashboard-mounted monitoring lamp may be in any color desired.

The flashing frequency is 90 ± 30 cycles per minute with a relative illumination period of 30 to 80%. Light must be emitted within 1.5 s of lamp switch-on. All the turn-signal lamps on one side of the vehicle must flash synchronously. If one lamp fails, the remaining lamps must continue to generate visible light.

All the turn-signal lamps flash synchronously in hazard-warning mode, also operational when the vehicle is stopped. An operation indicator is mandatory.

Two lamps each (color: yellow) are prescribed for the front, the rear and the side turn-signal lamps. In the USA the color red or yellow is permitted for the rear and side turn-signal lamps (SAE J588, Nov. 1984, [33]).

Figure 29: Positioning of front turn-signal lamps as per ECE

Dimensions in mm.

1) Less than 2,100 mm if vehicle body type prohibits compliance with regulations on maximum height.

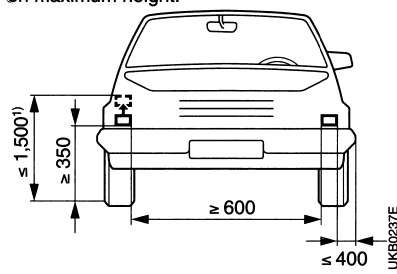


Figure 30: Positioning of rear turn-signal lamps as per ECE

Dimensions in mm Height and

width same as front turn-signal lamp.

1 Tail lamp.

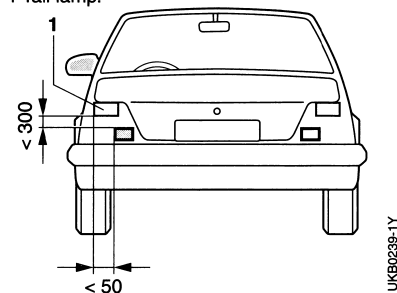


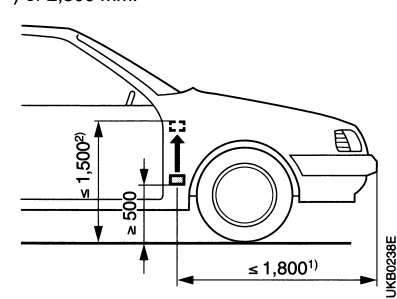
Figure 31: Positioning of side turn-signal lamps as per ECE

Dimensions in mm.

If the type of vehicle body does not permit adherence to the maximum dimensions:

1) or 2,500 mm.

2) or 2,300 mm.



Design regulations

The design requirements for Europe are set out in the ECE Regulations R6, R7, R23, R38 and R87 [6, 7, 9, 11, 16], and the installation requirements in ECE R48 [12] (Figures 29, 30 and 31).

In the USA, FMVSS 108 specifies the number, location, and color of signal lamps. The design and technical lighting requirements are defined in the relevant SAE standards.

Hazard-warning and turn-signal flashers for vehicles without trailer

The electronic hazard-warning and turn-signal flashers include a pulse generator designed to switch on the lamps via a relay and a current-controlled monitoring circuit to modify the flashing frequency in response to bulb failure. The turn-signal control stalk controls the turn signals, whereas the hazard flashers are operated using a separate switch.

Hazard-warning and turn-signal flashers for vehicles with trailer

This type of hazard-warning and turn-signal flasher differs from those employed on vehicles without trailers in the way that the function of the turn-signal lamps is controlled when they flash to indicate a change in direction.

Single-circuit monitoring

The tractor and trailer share a single monitoring circuit designed to activate the two indicator lamps at the flashing frequency. This type of unit cannot be used to localize lamp malfunctions. The flashing frequency remains constant.

Dual-circuit monitoring

Tractor and trailer are equipped with separate monitoring circuits. The malfunction is localized by means of the indicator lamp. The flashing frequency remains constant.

Tail and position lamps

(as per ECE R7 [7])

According to ECE R48 and 76/756/EEC, vehicle and trailer combinations wider than 1,600 mm require position lamps (facing forwards). Tail lamps (at rear) are mandatory equipment on vehicles of all widths. Vehicles wider than 2,100 mm

(e.g. trucks) must also be equipped with clearance lamps visible from the front and rear.

Position lamps

Two white-light position lamps are stipulated. The regulations set out in SAE J222, Dec. 1970, [29] apply in the USA.

Tail lamps

Two red tail lamps are stipulated. When the tail and stop lamps are combined in a nested design, the luminous-intensity ratio for the individual functions must be at least 1:5. Tail lamps must operate together with the position lamps.

The regulations set out in SAE J585, Feb. 2008, [30] apply in the USA.

Clearance lamps

(as per ECE R7 [7]).

Vehicles wider than 2,100 mm require two white lamps facing forward, and two red lamps facing to the rear. They must be positioned as far outward and as high as possible.

The regulations set out in SAE J592e, [34] apply in the USA.

Side-marker lamps

(as per ECE R91 [17]).

According to ECE R48, vehicles of any length exceeding 6 m must have yellow side-marker lamps (SML) except on vehicles with cab and chassis only.

Type SM1 side-marker lamps may be used on vehicles of all categories; type SM2 side-marker lamps, on the other hand, may only be used on cars.

The regulations set out in SAE J592e] apply in the USA.

Rear reflectors

(as per ECE R3 [4]).

According to ECE R48, two red, non-triangular, rear reflectors are required on motor vehicles (one on motorcycles and motorbicycles).

Additional reflective items (red reflective tape) are permitted if they do not impair the function of the legally required lighting and signaling equipment.

Two colorless, non-triangular front reflectors are required on trailers and on vehicles on which all forward-facing lamps with reflectors are concealed (e.g. retractable headlamps). They are permitted on all other types of vehicle.

Yellow, non-triangular, side reflectors are required on all vehicles with a length exceeding 6 m, and on all trailers. These are permitted on vehicles shorter than 6 m.

Two red triangular rear reflectors are required on trailers, but are banned on motor vehicles.

There may be no light fitted inside the triangle.

The regulations set out in SAE J594, Feb. 2010, [36] apply in the USA.

Parking lamps

(as per ECE R77 [15]).

ECE R48 permits either two parking lamps at the front and rear, or one parking lamp on each side. The prescribed colors are white facing forwards and red facing rearwards. Yellow may also be used at the rear if the parking lamps have been designed as single units together with the side turn-signal lamps.

The parking lamps must be designed to operate even when no other vehicle lights (headlamps) are on. The parking-lamp function is usually assumed by the tail and position lamps.

The regulations set out in SAE J222, Dec. 1970, [29] apply in the USA.

License-plate lamps

(as per ECE R4 [5]).

According to ECE R48, the rear license plate must be illuminated so as to be legible at a distance of 25 m at night.

Across the complete license-plate, luminance must be at least 2.5 cd/m^2 . The luminance gradient of $2 \times B_{\min}/\text{cm}$ should not be exceeded between any of the test points distributed across the surface of the license plate. $B_{\min}$ is defined as the smallest luminance measured at the test point.

The regulations set out in SAE J587, Oct. 1981, [32] apply in the USA.

As an alternative to the license-plate lamp, self-illuminating license plates will also be permitted in future.

Stop lamps

(as per ECE R7 [7]).

According to ECE R48, all cars must be fitted with two type S1 or S2 stop lamps, and one type S3 brake lamp, colored red in each case (Figure 32).

When a nested design with stop and tail lamps is used, the luminous-intensity ratio between individual functions must be at least 5 : 1.

The Category S3 brake lamp (central high-level brake lamp) must not be a combined unit incorporating any other lamp.

The regulations set out in SAE J586, Feb. 1984, [31] and SAE J186, Nov. 1982, [28] apply in the USA.

Figure 32: Positioning of stop lamps as per ECE

Dimensions in mm

1 Central high-level brake lamp (Category S3),

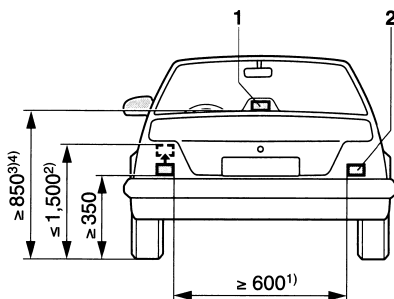
2 Two stop lamps (categories S1 and S2).

¹⁾ $\geq 400 \text{ mm}$ if width $< 1,300 \text{ mm}$,

²⁾ $\leq 2,100 \text{ mm}$ if compliance with max. height not possible, or

³⁾ $\leq 150 \text{ mm}$ below bottom edge of rear window,

⁴⁾ However, the lower edge of the central high-level brake lamp must be higher than the upper edge of the main stop lamps.



Rear fog warning lamps

(as per ECE R38 [11]).
For EU/ECE countries, ECE R48 pre-scribes one or two red rear fog warning lamps for all new vehicles. They must be distanced at least 100 mm from the brake lamp (Figure 33).

The visible illuminated area along the reference axis may not exceed 140 cm². The circuit must be designed to ensure that the rear fog warning lamp operates only in conjunction with the low beam, high beam and/or front fog lamp. It must also be possible to switch off the fog warning lamps independently of the front fog lamps.

Rear fog warning lamps may only be used if the visual range due to fog < 50 m, as rear fog warning lamps have a high luminous intensity that can severely dazzle drivers at the rear when there is a clear view. The required indicator lamp is yellow in color.

Reversing (backup) lamps

(as per ECE R23 [9]).
According to ECE R48, one or two white reversing lamps are permitted (Figure 34). The switching circuit must be designed to ensure that the reversing lamps operate only when reverse gear is engaged and the ignition on.

The regulations set out in SAE J593c, Feb. 1968, [35] apply in the USA.

Daytime running lamps and daytime driving lights

In the vehicle type approval the fitting and use of daytime running lamps has according to ECE R87 ([16], Figure 35) been mandatory in Europe since February 2011 for passenger cars and light commercial vehicles; since August 2012 this has also applied to all other vehicle categories.

Daytime running lamps must be automatically switched on on the vehicle when the engine is started and remain lit until either the front headlamps or fog lamps are switched on or the engine is turned off.

The regulations set out in FMVSS108 [27] apply in the USA.

Front cornering lamps

(as per ECE R119 [22]).
Two cornering lamps are permitted on the front of the vehicle which must radiate light

Figure 34: Positioning of reversing lamps as per ECE
Dimensions in mm.
Quantity: 1 or 2.

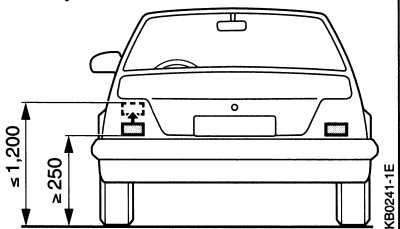
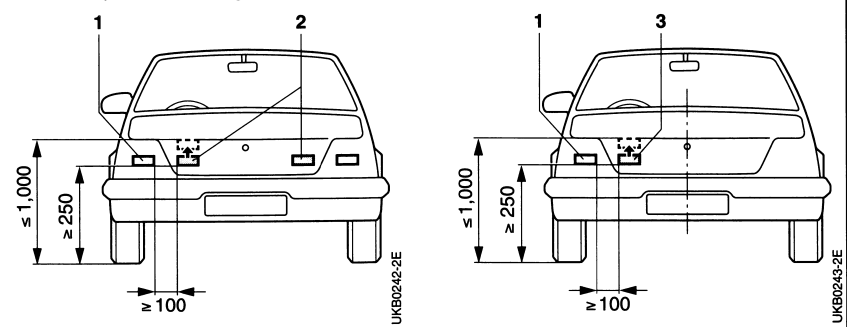


Figure 33: Positioning of rear fog warning lamps as per ECE

Dimensions in mm.
1 Brake lamp, 2 Two rear fog warning lamps, 3 One rear fog warning lamp for driving on right.



at least 60° to the outside of the vehicle. They are used in cornering maneuvers up to a maximum speed of 40 km/h to illuminate better the destination road (e.g. side roads or garage entrances) which is usually inadequately lit by the regular headlamps. The front cornering lamps are activated via the turn-signal control stalk or when the steering wheel is turned.

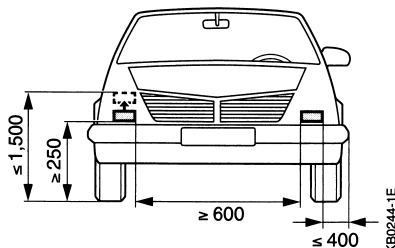
Front cornering lamps typically have the same light distribution as static cornering headlamps. The switching conditions, however, are different. For the most part the same reflector is used for both the static cornering headlamps and the front cornering lamps.

The regulations set out in SAE J852, Apr. 2001, [37] apply in the USA.

Identification lamps

According to ECE R65 [14], identification lamps must be visible from any direction, and create the impression of flashing. The flashing frequency is 2 to 5 Hz. Blue identification lamps are intended for installation on official vehicles. Yellow identification lamps are designed to warn of dangers or the transport of dangerous freight.

Figure 35: Positioning of daytime running lamps as per ECE
Dimensions in mm.



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Technical design variations for lamps

Light color

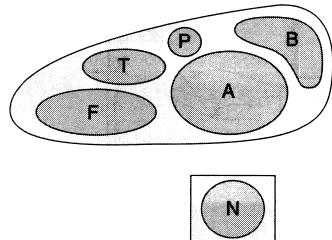
Depending on their application (e.g. stop lamps, turn-signal lamps, or rear fog warning lamps), motor-vehicle lamps must display uniform, distinctive colors in the red or yellow color range. These colors are defined in specific ranges of a standardized color scale (color location).

Since white light is composed of various colors, filters can be used to weaken or filter out the emission of undesirable spectral ranges (colors) completely. The color-filter functions may be performed by either the tinted lenses of the lamp, or the colored glass bulb of the lamp (e.g. yellow bulb in turn-signal lamps with clear lens).

Filter technology can also be used to design lamp lenses in such a way that, when the lamp is switched off, the color is matched to the vehicle's paintwork. Nevertheless, existing homologation regulations are complied with when the lamp is on. Color locations have been laid down in the EU/ECE region. They correspond to a wavelength of approximately 592 nm for "yellow/orange" turn-signal lamps, for example, and to a wavelength of approximately 625 nm for "red" brake and tail lamps.

Figure 36: Headlamps and lights at the vehicle front end

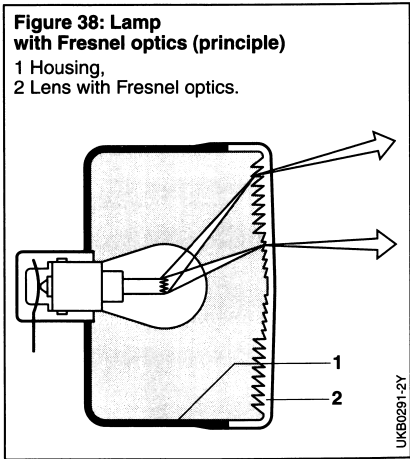
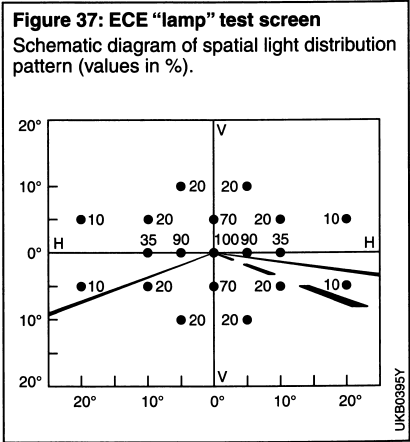
A Low-beam headlamp,
F High-beam headlamp,
T Daytime running lamp,
P Position/parking lamp,
B Turn-signal lamp,
N Fog lamp (optional).
T and P are often combined.



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Luminous intensities

As projected along different axes, minimum and maximum luminous intensities for all lamps must remain within a range calculated to guarantee signal recognition without, however, causing glare nuisance for other road users. The same percentile basis ("unified spatial light distribution pattern") is used for most lamps. With regard to the value along the reference axis, luminous-intensity levels to the sides and above and below may be lower (Figure 37). However, these values may, depending on the installation height or for special lamps (e.g. daytime running lamps), differ from this basis.



Lamps with Fresnel optics

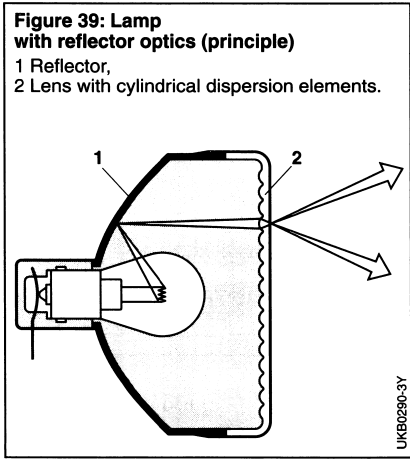
The light from the bulb is projected directly (without diversion) on the lens, which uses Fresnel optics to shape the beam in the desired manner (Figure 38). This concept is a highly cost-effective solution in that no vapor-deposited surface is needed. The drawbacks are low efficiency and the limited vehicle-design options.

Lamps with reflector optics

Lamps with approximately parabolic reflectors or stepped reflectors direct the light from the bulb in an axial beam, and shape the beam by means of the optical dispersion elements in the lens (Figure 39).

Lamps with free-form reflectors achieve the required beam spread or light pattern, completely or partially, by directing the light by means of a reflector. The outer lens can thus be designed as a clear lens, or supplemented by cylindrical lenses in the horizontal or vertical direction.

Designs combining both the above principles have also been successfully employed. The principle of the free-form lamp with Fresnel cap combines excellent luminous efficiency with a variety of possible stylistic implementations. It is essentially the reflector that shapes the light pattern. The Fresnel cap improves luminous efficiency by diverting a proportion of the light, which would not



otherwise contribute to the function of the lamp, in the desired direction (Figure 40).

Mainly lamps with free-form reflectors are used in modern vehicle designs. These lamps offer the best opportunities of adaptation to the body shape and therefore to the available structural space, and at the same time of meeting stylistic requirements.

Figure 40: Free-form lamp with Fresnel cap (principle)

- 1 Reflector,
- 2 Fresnel cap,
- 3 Clear lens.

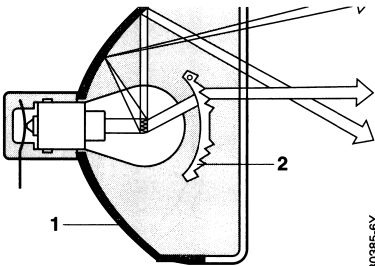
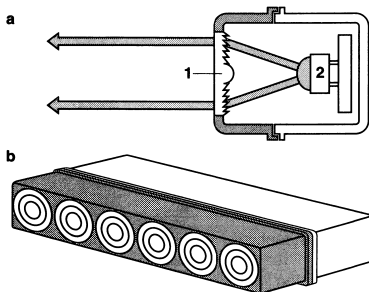


Figure 41: LED lamp with Fresnel optics (principle)

- a) Vertical section,
 - b) Overhead view.
- 1 Fresnel optics, 2 LED.



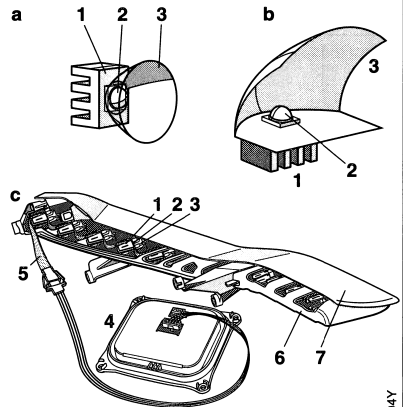
Lamps with light-emitting diodes

Light-emitting diodes (LEDs) are increasingly used as the light source for lamps. LEDs have been used for years as standard in auxiliary stop lamps. The crucial factor here is the design freedom to deliver a slimline design with a number of light sources (Figure 41 and Figure 42). In addition, LEDs in stop lamps provide an added safety aspect. An LED reaches maximum light output in less than 1 ms, whereas filament bulbs require about 200 ms to reach their nominal luminous flux. This means that LEDs emit the brake signal sooner, and this in turn reduces reaction time.

With their improved efficiency with regard to luminous flux, luminance, thermal performance and mechanical design, LEDs can also be used for functions with higher lighting requirements.

Figure 42: LED lamp

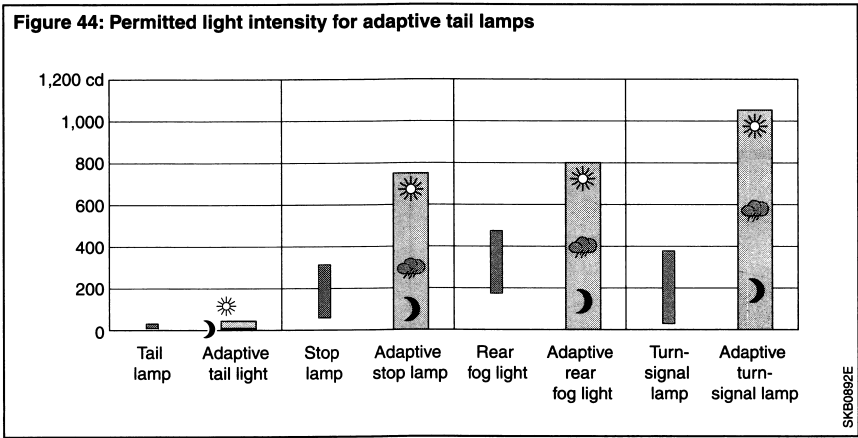
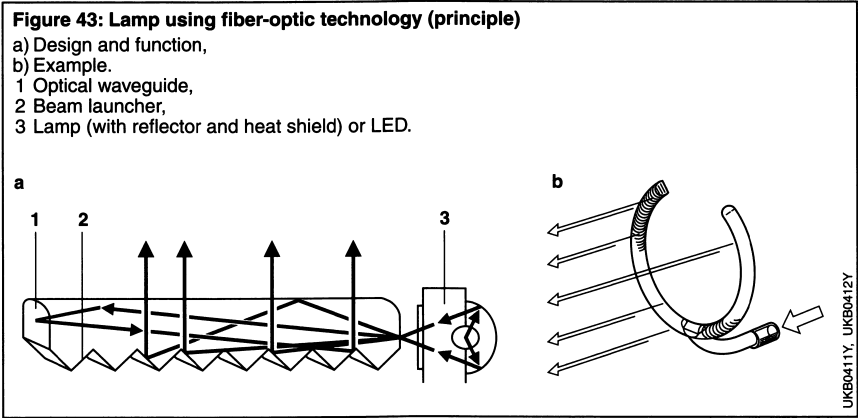
- a) With attachment optics,
 - b) With reflector,
 - c) Example of a daytime running lamp.
- 1 Heat sink,
 - 2 LED,
 - 3 Optics,
 - 4 ECU,
 - 5 Flexboard,
 - 6 Holder,
 - 7 Cover.



When it comes to being used as light sources in vehicle lights, LEDs differ crucially from filament and halogen lamps in that they cannot be operated directly from the vehicle electrical system. They require both a defined voltage, which ranges depending on the semiconductor material between 2.2 V and 3.6 V, and a defined current, by means of which the luminous intensity is adjusted. Resistor solutions can be used for activating functions with very low photometric requirements. Most applications require linear controllers or DC/DC converters. The electronics can be either integrated as individual electronics in the lamp or in a self-contained ECU, which is either mounted to the lamp or in the vehicle.

Lamps with fiber-optic technology

By using optical waveguides, the light source can be separated from the light emission point. In order to obtain the desired light pattern, special beam launchers in the optical waveguides, or optically active elements on or in front of the optical waveguide are required. Filament lamps can be used as light sources, but the high infrared levels are a drawback of these lamps. This necessitates the use of heat-resistant materials such as glass or heat shields. When using LEDs as “cold” light sources, it is possible to launch directly into transparent plastics such as PC (polycarbonate) or PMMA (polymethyl methacrylate). Lamps with fiber-optic technology are used primarily to produce



stylistic elements such as narrow strips or thin rings (Figure 43).

Adaptive rear-lighting systems

Until now, rear-lighting functions were produced with a one-level circuit. Depending on the version and design, this resulted in a fixed luminous intensity that had to lie within the legal limit values to guarantee minimum visibility.

A large number of sensors (brightness, dirt, visual range, wetness, etc.) mean that the vehicle is nowadays able to ascertain the ambient parameters and lighting conditions more exactly. So that the optimal visibility (e.g. sufficient luminous intensity without excessive glare) can be achieved, rear light functions will be able to vary the luminous intensity they emit depending on the ambient conditions detected around the vehicle. For example, a brake lamp can be operated with high luminous intensity during strong sunlight and with lower values at night in order to ensure optimal detection and attribution of the driver's actions (Figure 44).

Convenience lamps

Lights for use when the vehicle is stationary are increasingly being fitted. Typical applications are the illumination of areas immediately outside the doors when the doors are open or closed, lights that identify the vehicle outline in the underbody area, or lights which identify the door handles.

Combinations of such lights with position lamps or fog lamps are referred to as "coming home" functions, which are activated when the doors are unlocked, for instance.

Headlamp leveling control

Headlamp adjustment for low and high beams

The correct adjustment of vehicle headlamps is an essential factor in nighttime road safety for both the driver of the vehicle concerned and for oncoming vehicles. If the beam is set only a fraction too low, there will be a substantial reduction in headlamp geometric range (Table 4). If the beam is set only a fraction too high, oncoming traffic will suffer much greater glare.

Installation and regulations

Headlamp leveling, Europe

Since 1/1/1998, an automatic or manual headlamp leveling control (beam-height adjustment) device has been mandatory in Europe for all first-time vehicle registrations, except in cases where other equipment (e.g. hydraulic suspension leveling) guarantees that light-beam inclination will remain within the prescribed tolerances. Although this equipment is not mandatory in other countries, its use is permitted.

Automatic headlamp leveling control must be designed to compensate for vehicle laden states by lowering or raising the low beam by between 5 cm/10 m (0.5%) and 25 cm/10 m (2.5%).

A manual headlamp leveling control device is operated from the driver's seat and must incorporate a detent position in the base setting; beam adjustment is also performed at this position. Units with infinitely variable and graduated control must both feature visible markings in the vicinity of the hand switch for vehicle load conditions that require vertical aiming.

Table 4. Geometric range of the horizontal component of the low beam's light-dark boundary
Headlamp installed at height of 65 cm.

Inclination of the light-dark boundary (1% = 10 cm/10 m)	%	1.0	1.5	2.0	2.5	3.0
Adjustment dimension <i>e</i>	cm	10	15	20	25	30
Geometric range for the horizontal portion of the light-dark boundary	m	65.0	43.3	32.5	26.0	21.7

Technical design variations

All design variations employ an adjustment mechanism to provide vertical adjustment of the headlamp reflector (housing design) (Figure 45). Manually operated units employ a switch near the driver's seat to control the setting (Figure 46), while automatic units rely on level sensors on the vehicle axles to monitor suspension spring compression, and relay the proportional signals to the adjustment mechanisms.

Hydromechanical systems

Hydromechanical systems operate by transmitting a fluid through the connecting hoses between the hand switch (or level sensor) and the adjustment mechanisms. The degree of adjustment corresponds to the quantity of pumped fluid.

Vacuum systems

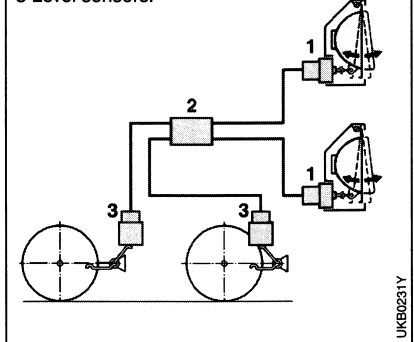
With vacuum systems, the hand switch (or level sensor) regulates vacuum from the intake manifold, and transmits it to the adjustment mechanisms to achieve varying degrees of adjustment.

Electrical systems

Electric systems employ electric gear motors as the adjustment mechanisms. They are switched either by switches in the vehicle or by axle sensors.

Figure 45: Automatic headlamp leveling control (principle)

- 1 Adjustment mechanism,
- 2 Processing unit,
- 3 Level sensors.



Headlamp adjustment

Headlamp adjustment, Europe

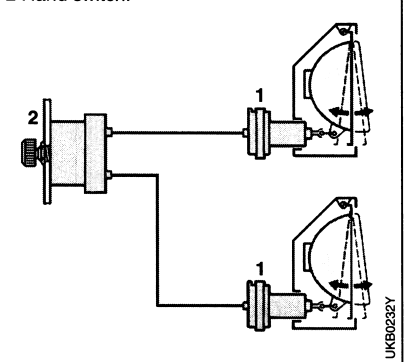
Regulations and procedure

Correct adjustment of motor-vehicle headlamps should ensure the best possible illumination of the roadway at the same time as minimizing glare for other traffic. The EU Directives (and therefore the requirements of Article 50 of the StVZO [2] (Road Traffic Licensing Regulations) in Germany) specify the required horizontal and vertical alignment of the headlamp beams. Low-beam glare is regarded as eliminated if the luminous intensity is not more than 1 lux at a distance of 25m in front of each headlamp when the beam is projected on a surface vertical to the road surface at the height of the center of the headlamp and above. If, however, the vehicle is subject to extreme variations in attitude due to changes in payload, the headlamps must be adjusted so that the desired aim is achieved.

EECE R48 [12] and EEC 76/756 [24] define the basic setting and the adjustment dimension required on the vehicle. For vehicle categories not covered by these directives, the regulations contained in ECE R48 or ECE R53 [13] shall apply.

Figure 46: Manual headlamp leveling (principle)

- 1 Adjustment mechanism,
- 2 Hand switch.



*Preparations for adjustment**Laden state of vehicle*

- Motor vehicles excluding motorcycles: unladen, 75 kg (one person in driver's seat)
- Motorcycles: for EC (as per 93/92/ EEC [25]) unladen, without a person on the driver's seat; for ECE and StVZO (Road Traffic Licensing Regulations) unladen, 75 kg (one person on the driving seat).

Suspension

- Vehicles without self-leveling suspension are rolled for few meters or rocked so that the suspension settles to a level position.
- Vehicles with self-leveling suspension are set and operated to the correct level according to the operating instructions.

Tire pressure

The tire pressure must be adjusted as per the vehicle manufacturer's instructions in accordance with the laden state.

*Compulsory function checks**on headlamp-range adjustment systems*

- Systems that operate automatically should be set or operated according to the manufacturer's instructions.
- On vehicles registered before 1/1/1990, a positively locating adjuster position is not required.
- Manually operated systems with two positively locating adjuster positions: On vehicles on which the headlamp beam is raised as the vehicle payload is increased, the adjuster must be set to the position at which the beam is at its highest (smallest degree of dip). On vehicles on which the headlamp beam is lowered as the vehicle payload is increased, the adjuster must be set to the position at which the beam is at its lowest (smallest degree of dip).

Surface and testing environment

- The vehicle and headlamp tester must be standing on a flat surface (based on ISO 10604 [26]).
- Adjustments and tests should be carried out in an enclosed space where the lighting conditions are not too bright.

*Adjusting and testing**with a headlamp aiming device*

- The headlamp aiming device should be aligned at the specified distance in front of the headlamp under test (except in the case of automatic equipment).
- In the case of headlamp aiming devices which do not run on tracks, vertical alignment relative to the vehicle's longitudinal center axis should be carried out separately for each headlamp. The test should then be carried out without moving the unit sideways again.
- In the case of headlamp aiming devices which run on tracks (or the like), alignment with the vehicle's longitudinal center axis need only be carried out once at the most favorable position (e.g. in a central position in front of the vehicle).
- The specified adjustment dimension for the headlamp concerned should then be set on the headlamp aiming device, and the headlamp adjustment checked and the headlamp adjusted to the correct setting.

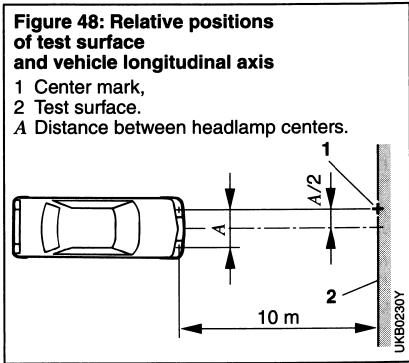
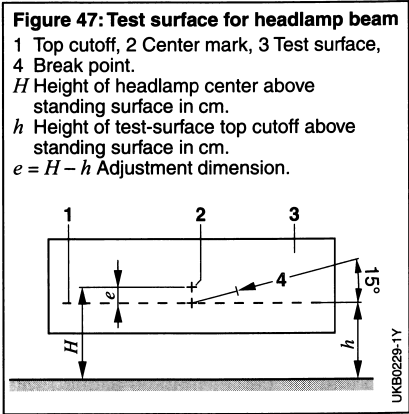
Adjusting and testing with a test surface

- The test surface must be vertical to the surface on which the vehicle is standing and at right angles to the vehicle's longitudinal center axis.
- The test surface should be finished in a light color, adjustable vertically and horizontally, and marked with the markings shown in Figure 47.
- The test surface should be positioned at a distance of 10 m in front of the vehicle so that the center mark is aligned with the center of the headlamp under test or adjustment (Figure 48). In the case of lamps with very low beams (e.g. fog lamps), a shorter distance can be chosen by calculating the adjustment dimension accordingly.
- Each headlamp must be set individually. Therefore, the other headlamp(s) must be covered over.
- The vertical position of the test surface must be set so that the top cutoff (parallel to the ground) is at the height $h = H - e$. If the surface is not 10 m from the vehicle, the setting e must be converted accordingly.

Notes on adjustment

In the case of headlamps with asymmetrical lower beams and fog lamps, the highest position of the light-dark boundary must be on the top cutoff and run as horizontally as possible across the minimum width of the test surface. The lateral adjustment of the headlamps must be such that the light pattern is positioned as symmetrically as possible about the perpendicular line running through the center mark.

In the case of headlamps for asymmetrical lower beam, the light-dark boundary must touch the top cutoff to the left of the center. The intersection between the left section (which should be as horizontal as possible) and the sloping section on the right of the light-dark boundary must be located at the perpendicular line passing through the center mark.



The center of the high beam must be on the center mark.

For headlamps which feature a common adjustment facility for low beam and fog lamps or for high beam, low beam and fog lamps, it is always necessary to use the low-beam headlamp as the basis for adjustment (adjustment dimension e , see Table 5).

Headlamp aiming devices

Function

Correct adjustment of motor-vehicle headlamps should ensure the best possible illumination of the roadway by the low beam while minimizing glare for oncoming traffic at the same time. For this purpose, the inclination of the headlamp beams with respect to a level base surface, and their direction to the vertical longitudinal center plane of the vehicle, must satisfy official directives.

Equipment design

Headlamp aiming devices are portable imaging chambers (Figure 49). They comprise a single lens and an aiming screen located in the focal plane of the lens, and are rigidly connected to it. The aiming screen has markings to facilitate correct headlamp adjustment, and can be viewed by the equipment operator using suitable devices such as windows and adjustable

Table 5: Headlamp adjustment (excerpt from StVZO (Road Traffic Licensing Regulations))

Vehicle type: Motor veh., > 2 whs. Headlamp position: height above road surface	Adj. dimension "e"	
	Dipped beam	Fog beam
Approved to 76/756/EEC or ECE R48 and StVZO, first registered on 1/1/1990 or later, < 1,200 mm	Setting on vehicle e.g. $\varnothing 10\%$	-2.0 %
First registered 12/31/1989 or earlier, $\leq 1,400$ mm, and first registered after 12/31/1989, > 1,200 mm, but $\leq 1,400$ mm	-1.2 %	-2.0 %

refraction mirrors. The prescribed headlamp adjustment dimension e , i.e. the inclination relative to the centerline of the headlamp in cm at a fixed distance of 10 m, is set by turning a knob to move the aiming screen (Tables 4 and 5).

The aiming device is aligned with the vehicle axis using a sighting device, such as a mirror with an orientation line. It is turned and aligned so that the orientation line uniformly touches two external vehicle-reference marks. The imaging chamber can be moved vertically and clamped at the level of the vehicle headlamp.

Headlamp testing

The headlamp can be tested after the equipment has been correctly positioned at the front of the lens. An image of the light pattern emitted by the headlamp appears on the aiming screen. Some test devices are also equipped with photodiodes and a display to measure the luminous intensity.

On headlamps with asymmetrical lower-beam patterns, the light-dark boundary should touch the horizontal top cutoff; the intersection between the horizontal and sloping sections must be located on the perpendicular line running through the center mark (Figure 50). After adjust-

ing the lower-beam light-dark boundary in accordance with the regulations, the center of the upper beam (assuming that high and low beam are adjusted together) should be located within the rectangular border about the center mark.

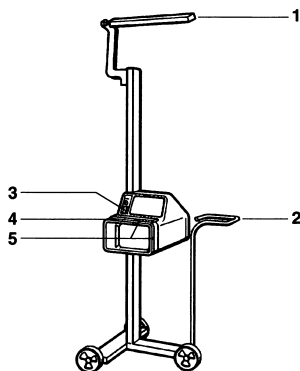
Headlamp adjustment, USA

For headlamps compliant with US Federal legislation, visual (vertical only) adjustment as permitted since 5/1/1997 has become increasingly widespread in the US since mid-1997. There is no horizontal aiming here.

Before then, the use of mechanical aiming devices was the most common method of headlamp adjustment in the USA. The headlamp units were equipped with three pads on the lenses – one for each of the three adjustment planes. A calibrating unit is placed against these pads. Aiming is checked using spirit levels. With the VHAD (Vehicle Headlamp Aiming Device) method permitted since 1993, the headlamps are adjusted relative to a fixed vehicle reference axis. This procedure is carried out using a spirit level firmly attached to the headlamp. As a result, the three lens pads were no longer required.

Figure 49: Aiming device for headlamps

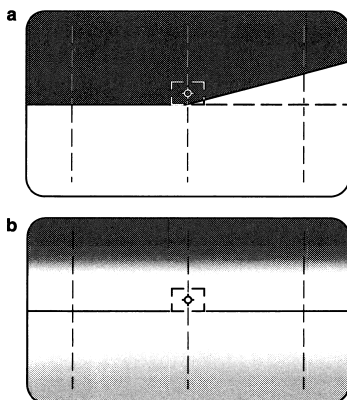
- 1 Alignment mirror,
- 2 Handle,
- 3 Luxmeter,
- 4 Refraction mirror,
- 5 Markings for center of lens.



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Figure 50: Viewing window in aiming device

- a) Top cutoff for light-dark boundary for asymmetrical lower beam,
- b) Center mark for middle of upper-beam pattern.



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References

[1] StVZO §22a: Design certification for vehicle parts.

[2] StVZO §50: Headlight for high and low beams.

[3] StVZO §52: Zusätzliche Scheinwerfer und Leuchten.

[4] ECE R3: Uniform provisions concerning the approval of retro-reflecting devices for power-driven vehicles and their trailers.

[5] ECE R4: Uniform provisions concerning the approval of devices for the illumination of rear registration plates of power-driven vehicles and their trailers.

[6] ECE R6: Uniform provisions concerning the approval of direction indicators for power-driven vehicles and their trailers.

[7] ECE R7: Uniform provisions concerning the approval of front and rear position (side) lamps, stop-lamps and end-outline marker lamps for motor vehicles (except motorcycles) and their trailers.

[8] ECE R19: Uniform provisions concerning the approval of motor vehicle front fog lamps.

[9] ECE R23: Uniform provision concerning the approval of reversing lamps for power-driven vehicles and their trailers.

[10] ECE R37: Uniform provisions concerning the approval of filament lamps for use in approved lamp units on power-driven vehicles and their trailers.

[11] ECE R38: Uniform provisions concerning the approval of rear fog lamps for power-driven vehicles and their trailers.

[12] ECE R48: Uniform provisions concerning the approval of vehicles with regard to the installation of lighting and light-signalling devices.

[13] ECE R53: Uniform provisions concerning the approval of category L3 vehicles with regard to the installation of lighting and light-signaling devices.

[14] ECE R65: Uniform provisions concerning the approval of special warning lamps for motor vehicles.

[15] ECE R77: Uniform provisions concerning the approval of parking lamps for power-driven vehicles.

[16] ECE R87: Uniform provisions concerning the approval of daytime running lamps for power-driven vehicles.

[17] ECE R91: Uniform provisions concerning the approval of side-marker lamps for motor vehicles and their trailers.

[18] ECE R98: Uniform provisions concerning the approval of motor vehicle headlamps with gas-discharge light sources.

[19] ECE R99: Uniform provisions concerning the approval of gas-discharge light sources for use in approved gas-discharge lamp units of power-driven vehicles.

- [20] ECE R112: Uniform provisions concerning the approval of motor vehicle headlamps emitting an asymmetrical passing beam or a driving beam or both and equipped with filament lamps and/or light-emitting diode (LED) modules.
- [21] ECE R113: Uniform provisions concerning the approval of motor vehicle headlamps emitting a symmetrical passing beam or a driving beam or both and equipped with filament lamps.
- [22] ECE R119: Uniform provisions concerning the approval of cornering lamps for power-driven vehicles.
- [23] ECE R123: Uniform provisions concerning the approval of adaptive front-lighting systems (AFS) for motor vehicles.
- [24] 76/756/EEC: Council Directive of 27 July 1976 on the approximation of the laws of the Member States relating to the installation of lighting and light-signaling devices for motor vehicles and their trailers.
- [25] 93/92/EEC: Council Directive of 29 October 1993 on the installation of lighting and light-signalling devices on two or three-wheeled motor vehicles.
- [26] ISO 10604: Road vehicles; measurement equipment for orientation of headlamp luminous beams.
- [27] FMVSS 108: Lamps, reflective devices, and associated equipment.
- [28] SAE J186: Supplemental High Mounted Stop and Rear Turn Signal Lamps for Use on Vehicles Less than 2032 mm in Overall Width.
- [29] SAE J222: Parking Lamps (Front Position Lamps).
- [30] SAE J585: Tail Lamps (Rear Position Light) for Use on Motor Vehicles Less than 2032 mm in Overall Width.
- [31] SAE J586: Stop Lamps for Use on Motor Vehicles Less Than 2032 mm in Overall Width.
- [32] SAE J587: License Plate Illumination Devices (Rear Registration Plate Illumination Devices).
- [33] SAE J588: Turn Signal Lamps for Use on Motor Vehicles Less Than 2032 mm in Overall Width.
- [34] SAE J592e: Clearance, Side Marker and Identification Lamps.
- [35] SAE J593c: Back-up Lamps.
- [36] SAE J594: Reflex Reflectors.
- [37] SAE J852: Front Cornering Lamps for Use on Motor Vehicles.
- [38] D. Meschede: Gerthsen Physik. 24th Edition, Springer-Verlag, 2010.
- [39] R. Baer: Beleuchtungstechnik – Grundlagen. 3rd Edition, Huss-Medien-GmbH, Verlag Technik, 2006.

Automotive glazing

The material properties of glass

Basic components

Window panes for automotive use are made of silica glass. The basic chemical constituents and their proportions are as follows:

- 70 to 72% silicic acid (SiO_2) as the basic component of glass
- Approx. 14% sodium oxide (Na_2O) as flux
- Approx. 10% calcium oxide (CaO) as stabilizer.

These substances are mixed in the form of quartz sand, soda ash, and limestone. Other oxides such as magnesium and aluminum oxide are added to the mixture in proportions of up to 5%. These additives improve the physical and chemical properties of glass.

Manufacturing flat glass

The glass panes are made out of the basic product, flat glass. Flat glass that is cast using the float-glass process is used. This process involves melting the mixture at a temperature of $1,560^\circ\text{C}$. The melt then passes through a refining zone at $1,500$ to $1,100^\circ\text{C}$, and is then floated on a float bath of molten tin. The molten glass is heated from above (smoothing of the surface by fire-finishing). The flat surface of the molten tin creates flat glass with flat parallel surfaces of a very high quality (tin bath underneath, fire-finishing on top). The glass is cooled to 600°C before it is lifted out of the float bath into the cooling section. After a further period of slow, non-stress cooling, the glass is cut into sheets measuring $6.10 \times 3.20 \text{ m}^2$.

Tin is suitable for the float-glass process because it is the only metal that does not produce any vapor pressure at $1,000^\circ\text{C}$ and is liquid at 600°C .

Table 1: Material properties and physical data of glass and finished windshields and windows

Property	Dimension	TSG	LSG
Density	kg/m^3	2,500	2,500
Hardness	Mohs	5 to 6	5 to 6
Resistance to pressure	MN/m^2	700 to 900	700 to 900
E module	MN/m^2	68,000	70,000
Flexural strength			
Before pre-tension	MN/m^2	30^2	30^1
After pre-tension	MN/m^2	50^2	
Specific heat	$\text{kJ/kg} \cdot \text{K}$	0.75 to 0.84	0.75 to 0.84
Thermal conduction coefficient	$\text{W/m} \cdot \text{K}$	0.70 to 0.87	0.70 to 0.87
Coefficient of thermal expansion	K^{-1}	$9.0 \cdot 10^{-6}$	$9.0 \cdot 10^{-6}$
Relative permittivity		7 to 8	7 to 8
Light transmittance (DIN 52306) [1] clear ³	%	≈ 90	$\approx 90^1$
Refractive index ³		1.52	1.52^1
Deviation angle of wedge ³	Arc minute	< 1.0 flat < 1.5 curved	≤ 1.0 flat ¹ ≤ 1.5 curved ¹
Dioptric divergence DIN 52305 [2] ³	Diopters	< 0.03	$\leq 0.031^1$
Thermal stability	$^\circ\text{C}$	200	90^1 (max. 30 min)
Resistance to temperature shocks	K	200	

¹ Properties of finished laminated safety glass (LSG).

In calculating the permissible bending stress, the coupling effect of PVB film is to be disregarded.

² Calculated values; these values already contain the necessary safety margins.

³ Figures for optical properties depend very heavily on the type of window.

Automotive windshield and window glass

Glass used for automotive glazing is of two types:

- Single-pane toughened safety glass (TSG), which is chiefly used for side-window, rear-window and sunroof glazing.
- Laminated safety glass (LSG), which is used primarily for windshields and rear windows, but also for sunroofs. LSG is also increasingly being fitted in vehicle side and rear windows.

The materials from which TSG and LSG panes are made are the basic glass types:

- Transparent float glass: This glass offers the best possible light transmittance.
- Tinted float glass: This glass has a homogeneous green or gray tint within the material; the tint blocks heat from the sun.
- Coated float glass: The glass is coated on one side with noble-metal and metal oxides; the coating reduces heat and UV radiation entering the vehicle, thus providing thermal insulation.

TSG panes

TSG panes differ from LSG panes as they have greater mechanical and thermal strength, and their breaking and shattering behavior is different. They pass through a toughening process which greatly prestresses the surface of the glass. In case of breakage, these panes shatter into many small blunt-edged pieces (to eliminate the risk of injury).

Post-processing (by grinding or drilling) of TSG panes is not possible. The standard thicknesses are 3, 4 and 5 mm.

LSG panes

An LSG pane is made of a crack-proof, flexible plastic intermediate layer of polyvinyl butyral (PVB) bonded between two sheets of glass. When subjected to impact or shock, the glass splits into web-like crack patterns. The plastic intermediate layer holds the broken pieces of glass together (to eliminate the risk of injury). The laminate retains its integrity and transparency when the glass is shattered.

The standard thicknesses for LSG panes are 4.5 to 5.6 mm.

Optical properties

The requirements for the optical quality of automotive glazing are as follows:

- unimpeded vision,
- flawless vision,
- undistorted vision.

Achievement of optimum optical quality has to be balanced against structural requirements and the vehicle-body design, taking account of such factors as

- glazing with large surface areas,
- glazing that is fitted laying flat,
- cylindrical or spherical panes,
- panes with a high degree of curvature.

Possible quality impairments arise from:

- optical deflection,
- optical distortion,
- double imaging.

Optical deflection increases with:

- increasing obliqueness of the angle of incidence, i.e. increasing slope of the window,
- increasing pane thickness,
- reducing radius of curvature (increasing degree of bend),
- increasing divergence from perfect surface parallelism of the original glass sheet.

Green- or gray-tinted glass is used as heat absorption glass because it blocks the transmission of infrared rays (heat radiation) more strongly than shorter wavelengths. On the other hand, it also reduces transmittance within the visual spectrum. The PVB film in LSG absorbs light in the ultraviolet range.

The optical properties of TSG and LSG panes are roughly the same because the optical properties of the intermediate plastic layer in LSG are very similar to those of glass in the visible spectrum.

Functional design glazing

The demands placed on glazing are continually increasing. The flat panes used in the past simply served to protect occupants from wind and weather. Now, automotive glazing performs a wide variety of functions.

Tinted glazing

This is made from glass which is tinted within the material, and reduces the direct penetration of solar radiation into the vehicle interior.

The reduction in the transmittance of solar energy occurs chiefly in the long-wavelength spectrum (infrared, Figure 1), and this mainly lowers the transmittance of energy, thus reducing the heat transmitted to the vehicle interior. The degree to which light transmittance within the visible spectrum is affected depends on the degree of color tinting and the thickness of the glass.

For windshields, light transmittance must be at least 75%. Strongly tinted glass with a light transmittance of less than 70% can be used in windows from

the B-pillar to the rear, if the vehicle has two exterior mirrors. For sunroof glazing, tinted glass with a significantly lower light transmittance and a UV transmittance of $\leq 2\%$ is used.

Coated glazing

This type of glazing uses glass with a metal or metal-oxide coating. Depending on the production process, the coating may be carried out before or after the sheet of glass is bent and prestressed. The coating is applied to the inner glass surface of an LSG pane.

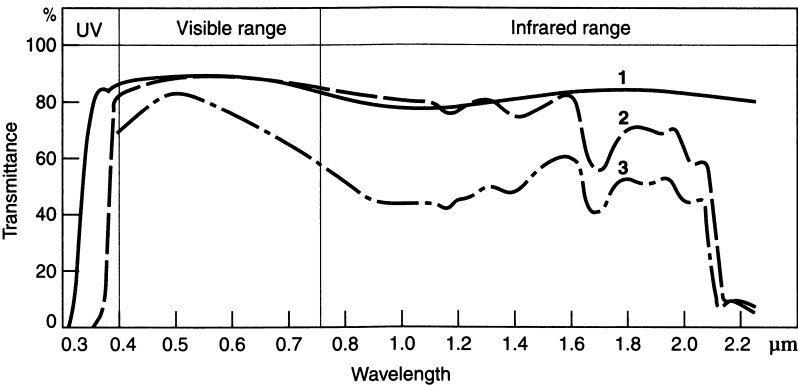
This type of coated glazing has a light transmittance of less than 70%. Consequently, it can be fitted from the B-pillar to the rear, if the vehicle has two exterior mirrors.

Coated glass can also be used for sunroof glazing. Another type of sunroof glazing uses pyrolytically coated windows which are post-processed after the coating has been applied.

Coated glazing reduces direct solar radiation into the vehicle interior and absorbs solar energy in the infrared and ultraviolet ranges.

Figure 1: Light transmittance of automotive windshield and glass

- 1 Float glass and TSG windows, thickness 4 mm, non-tinted,
- 2 LSG windows, overall thickness 5.5 mm, non-tinted,
- 3 LSG windows, overall thickness 5.5 mm, green.



Windshields with sunshield coating

A coating is applied to the inner surface of the outer or inner layer of glass in a laminated glass pane. The coating is a multi-layer interference system with silver as its base layer. As the coating is on the inside of the laminate, it is permanently protected against corrosion and scratching.

The purpose of the coating is to reduce the transmittance of solar energy by more than 50%, thus reducing the heat transmitted to the vehicle interior. Transmittance is reduced primarily in the infrared range so that visible light transmittance is altered negligibly. The reduction is achieved largely by reflection, so that secondary reflection into the interior is low. The light transmittance of this glass in the UV range is very low, i.e. less than 1%.

Sunroof glazing made from laminated safety glass

The bent laminated safety glass consists of two tinted-glass layers which are thermally partially prestressed in order to increase mechanical strength. They are bonded to either side of a highly crack-resistant and specially tinted film. The overall thickness depends on the surface area of the glass, and the overall design of the sunroof.

Absorption that takes place mainly in the infrared range guarantees minimum heat penetration. The coating also provides low light transmittance and complete filtering of UV light.

Automotive insulation glass

Automotive insulation glass is made of two flat or curved sheets of single-pane toughened safety glass (3 mm) separated by an air gap (3 mm). This type of insulation glass reduces heating of the vehicle interior, particularly in combination with a tinted coating. Transmittance is reduced primarily in the infrared range so that visible light transmittance is altered negligibly.

Insulation glass also provides better thermal insulation in the winter and improves sound insulation.

Insulation glass is now only used on commercial vehicles such as buses, trains and airplanes.

Heated laminated safety glass

Heated laminated safety glass can be used for windshields or rear windows. They prevent icing and misting of the glass pane, even in extreme winter temperatures, and ensure clear visibility.

Heated laminated safety glass is made up of two more sheets of glass with a PVB film bonded between them. Inside the PVB film, there are heater filaments which may be less than 20 μm thick, depending on the heat output required. The heater filaments may be laid in a waveform pattern or in straight lines. They may also run vertically or horizontally. The heated area may cover the whole of the pane or may be divided into separate zones with varying heat outputs. In this way, the windshield wipers can be prevented from freezing to the windshield in subzero temperatures.

Heated windows can also be created by layering.

Automotive antenna glass

This type of glass has antenna wire embedded in it. In the case of windows made of single-pane toughened safety glass (roof and side windows), the antenna is printed on the glass and is located – almost invisibly – on the inner surface of the window. With laminated safety glass windows (windshields), the antenna wiring system is embedded or printed onto the inner film.

Acoustic glazing

An acoustic windshield is composed, like every windshield, of two individual sheets of glass – an outer and an inner sheet which are firmly connected to each other by way of a sturdy plastic film of polyvinyl butyral (PVB). In the case of acoustic glazing, however, the conventional PVB film is replaced by acoustic PVB which has a highly damping acoustic core located in the middle between two layers of standard PVB. The two outer layers of film of conventional plastic thus safeguard the mechanical properties, while the core layer of damping material absorbs vibrations.

Acoustic windshields reduce the transmission of low-frequency engine vibrations into the vehicle interior. Engine-vibration noises are absorbed by the windshield, these booming noises thus being reduced at low frequencies by up to 5 dB.

Water-repellent glazing

A coating on the glass causes tiny pearls of water to form on the glass surface when it rains. These are blown off the glass by the air stream.

All glass surfaces can be fitted with this function. After several years of use, the effectiveness of water-repellent glazing can be fully restored with a regenerating kit.

Electrochromic glazing

The standard version of electrochromic glass consists of two 2.1 mm thick glass panes and a thin intermediate layer with an overall thickness of 5.8 mm and a weight of 16 kg/m². Depending on the size of the glass, the light transmittance can be changed entirely in 30 to 60 seconds. This reaction time remains constant over a temperature range of –25 to +90 °C. The glass remains transparent in every tint, i.e. it never becomes opaque. A ± 1.5 V DC power supply is required to activate the glass. Power consumption is extremely low at less than 0.1 Wh/cycle/m².

Preferred areas of application for electrochromic glazing are glass roofs, rear windows and triangular windows from the C-pillar.

Panorama roofs

There is a wide selection of panorama roofs available, ranging from the closed, large-surface glass roof, through the twin- or multi-panel roof, to the lamellar sunroof.

The following glass technologies can be used for panorama roofs:

- single-pane safety glass, 5 mm or 4 mm thick,
- conventional laminated safety glass, 6 mm and 5 mm thick,
- partially prestressed laminated safety glass, 5 mm thick.

References

- [1] DIN 52306: Ball drop test on safety glass for vehicle glazing.
- [2] DIN 52305: Determination of the optical deviation and refractive power of safety glass for vehicle glazing.

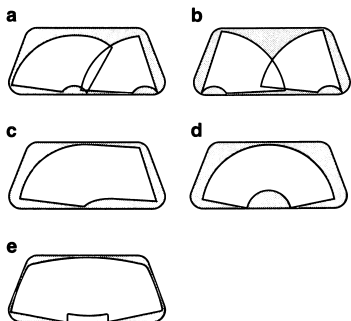
Windshield, rear-window and headlamp cleaning systems

The function of systems for cleaning the windshield and rear window is to provide the driver with sufficient visibility in the motor vehicle at all times. The following types of system are used:

- windshield wiper systems,
- rear-window wiper systems,
- washer systems in combination with wiper systems,
- headlamp washer systems.

Figure 1: Windshield cleaning systems

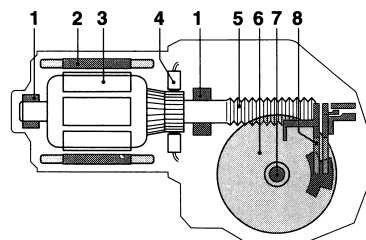
- a) Tandem system,
- b) Opposed-pattern system,
- c) Wiper-blade-controlled single-arm system,
- d) Non-controlled single-arm system (rear wiper),
- e) Stroke-controlled single-arm system.



UKW0252-3Y

Figure 2: Wiper drive with helical-gear mechanism

- 1 Bearing, 2 Magnet, 3 Armature,
- 4 Commutator with carbon brushes,
- 5 Worm, 6 Helical gear, 7 Output shaft,
- 8 Parking-position system.



UKW0253-2Y

Windshield wiper systems

Function and requirements

The function of wiper systems is to remove water, snow and dirt (mineral, organic or biological) from the windshield and rear window. Boundary conditions are:

- operational at high temperature (+80 °C) and low temperature (−40 °C),
- corrosion resistance against acids, bases, salts, ozone,
- endurance strength under load (e.g. due to high snow load),
- pedestrian protection,
- cleaning effect at high driving speeds,
- low operating noise.

Legislative bodies have translated the demand for adequate visibility into standardized areas of vision on the windshield (e.g. EEC for Europe [1] and FMVSS for the USA [2]). These are subdivided into several areas and must be cleaned by the wiper system to fixed percentage levels. The most important cleaning systems for passenger-car windshields which satisfy these requirements are shown in Figure 1.

Wiper systems for commercial vehicles are similar to those for passenger cars, but are subject to different requirements especially with regard to driving speed and windshield shape.

Drive

Windshield wiper systems consist of an electric motor with a helical-gear mechanism (drive), a linkage, the wiper bearings, the wiper arms, and the wiper blades.

Motor design

Permanent-magnet DC motors with integrated helical-gear mechanisms (Figure 2) are used to drive wiper systems. The motors are for cost, weight and space reasons high-speed in design. Adaptation to the speed and torque requirements of the wiper system is performed via the helical-gear mechanism.

The primary function of wiper drives is to guarantee visibility under anticipated conditions, i.e. sufficiently frequent wind-

shield wiping. For windshield wiper systems, this generally means roughly 40 times per minute and under more extreme conditions approximately 60 times. In the standard SAE J903 [3], this is laid down with the requirement of a speed setting of 45 and a speed jump of at least 15 wiping cycles per minute.

The classification by power output often used for electrical machines is not suitable for drives for wiper systems. Wiper systems are normally operated on a wet windshield; in this case, minimal torque is demanded of the motor at high rotational speeds. On the other hand, in the case of high-load states such as when the wiper blade is frozen, a high torque is required at low rotational speed. In both cases, the power output (product of torque and rotational speed) is low. Because the drive variable is dependent on the required torque at the high load point, this torque is used as the classification feature.

Conventional drives

Conventional drives (rotary drives) are characterized by the fact that the output shaft permanently rotates in the same direction. The actual wiping movement on the windshield is determined by the wiper linkage and its kinematic layout (Figure 3).

The different speed settings are achieved in such systems due to the fact that there is a third carbon brush connected to the battery voltage in addition to the carbon brushes connected to ground and to the battery voltage. This third carbon brush is at a defined angle to the other brushes and results through its commutation angle in a pull-out of the motor characteristic (Figure 4a).

The wiper drives have a sensor system to ensure that the wiper blades adopt the correct parking position. Because the position of the output shaft of the conventional drive has a 1:1 correlation with the wiping angle, this function can be guaranteed

Figure 3: Conventional (rotary) drive

a) Parallel drive, b) Series drive.

- 1 Motor crank, 2 Articulated rod,
- 3 Oscillating-crank link as ball pivot,
- 4 Oscillating crank,
- 5 Wiper bearing,
- 6 Wiper arm.

α Angle of motor rotation,

β, γ Wiping angle (β and γ can be different).
The arrows indicate the direction of motion.

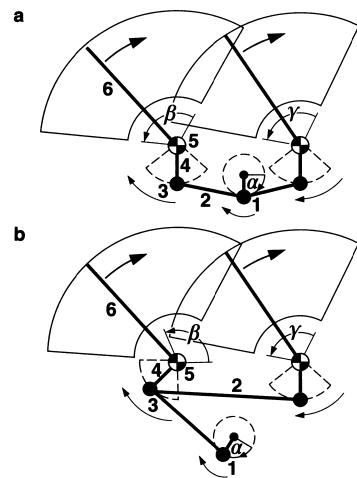
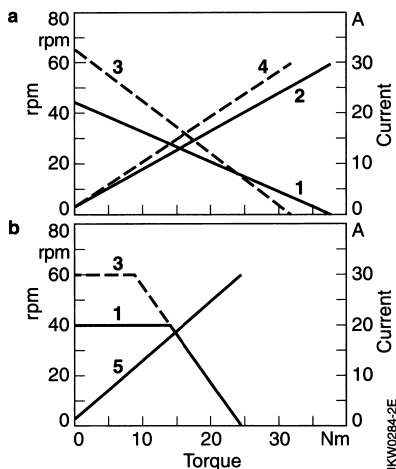


Figure 4: Drive characteristics of wiper drives

a) Conventional wiper drive,
b) Reversing wiper drive.

- 1 Speed setting 1,
- 2 Current setting 1,
- 3 Speed setting 2,
- 4 Current setting 2,
- 5 Current, reversing mode.



with a reference position at the output. Microswitches with cam activation, sliders with contact disks or Hall-effect sensors with a sensor magnet are used as the parking-position system.

Wiper drives are designed with high-load and anti-blocking protection. This protection can be provided conventionally by a thermostatic switch or by blocking or heavy-load detection in the triggering electronics.

Reversing drives

In the case of reversing drives, the output shaft oscillates at a defined angle, usually less than 180°. Output to the wiper shafts is provided, as is also the case with conventional systems, via a wiper linkage.

Reversing drives, unlike conventional rotary drives, have only two carbon brushes. The motor voltages required to effect the different wiping frequencies (Figure 4b) are provided via the integrated control electronics.

The position and speed of the output is relevant to controlling the drive. These variables are recorded by sensors (e.g. Hall-effect sensors with sensor magnet) and processed in the control electronics. The signals from this sensor system are usually used to detect the parking position and spare the parking-position system of conventional drives. High-load and blocking protection is likewise guaranteed by the integrated electronics.

Linkage

Function

The linkage is the connection between the drive and one or more wiper arms. It transfers the movement of the drive via the motor crank with ball pivot to the articulated rods and via these to the ball pivot of the wiper arm (Figure 3). The decisive variable for transfer is the wiping angle, i.e. the angle between the upper stop of the wiper blade close to the A-pillar and the lower stop close to the bottom edge of the windshield. This produces – in combination with the length of the wiper arms and the wiper blades and the position of the wiper bearings in relation to the windshield – the field of vision.

System concepts

With regard to the linkages, a distinction is made between purely mechanical wiper systems (rotary drives) and electronically controlled wiper systems based on reversing technology. The following list sets out the four most important concepts for current wiper systems.

- Rotary drive: The drive runs in a circle and the linkage transfers the circular motion into the oscillating motion of the wiper arms (Figure 3).
- Reversing technology: The electronically controlled drive rotates less than a half rotation. The linkage transfers the force from the drive to the wiper arms. The advantage of such systems over the rotary drive lies in the fact that the space required for the linkage kinematics is roughly halved.
- Two-motor wiper system: Each wiper arm is moved by its own drive with linkage.
- Direct wiper drive: The wiper arm is connected directly to the drive shaft; the linkage is omitted.

Attachment to the body

There are, depending on the wiper-system concept, different ways of attaching the system to the body. The wiping angle is influenced by the position of the drive in relation to the wiper arms and the position of the wiper bearings in relation to the vehicle. This angle in turn is one of the deciding variables for satisfying the field of vision required by the legislative bodies.

To simplify fitting on the vehicle and to limit the tolerances of the wiping angle, the wiper system is designed with a shaped tube or a shaped casting as a compact system, and secured to the drive and the wiper bearings.

As an alternative to this compact design, the drive and the bearings of the wiper arms can also be screwed directly to the body (loose-link connection); the shaped tube is omitted. This cuts down on the number of parts in the wiper system, but increases the amount of fitting work by the vehicle manufacturer in that more parts must be connected to the vehicle. Furthermore, this increases the demands placed on the rigidity of the vehicle body

and necessitates increased precision when installing the components in order to ensure wiping-angle precision.

Depending on the position of the drive in relation to the wiper bearings, either parallel coupling (i.e. both wiper arms are driven directly by the drive, Figure 3a) or series coupling (i.e. the drive moves only one wiper arm, while the second wiper arm is connected to the first arm, Figure 3b) is used.

In the case of two-motor wiper systems and the direct wiper drive, the drives are connected directly to the body.

It is important to optimize the linkage to ensure that the wiper system works harmoniously, i.e. the wiper blade moves uniformly over the windshield. Smooth operation is achieved and the reversing noise of the wiper blade on the windshield is reduced by matching the maximum values of angular acceleration and the force-transmission angles near to wiper-arm reversal points at the outer and lower edges of the windshield.

Large wiping angles or difficult transfer ratios give rise under certain circumstances to greatly changing wiper speed on the windshield in the edge areas, caused by the changing lever ratios. For this reason, crosslays are used in some vehicles (Figure 5).

The current trend towards weight reduction and associated CO₂ reduction is taken into account with electronically controlled wiper systems. Electronic control of the drives significantly reduces the space needed by the articulated rods of the linkage for movement. The installation space can be further reduced by the use of a two-motor system with two considerably smaller drives.

With the direct wiper drive, it is also possible to dispense with the linkage of the two-motor system.

Additional functions, such as, for example, an extended parking position, an overload-protection function (e.g. for snow), an infinitely variable wiping speed, and a consistently large wipe pattern over varying operating conditions (e.g. driving speed), can be implemented with electronically controlled wiper systems.

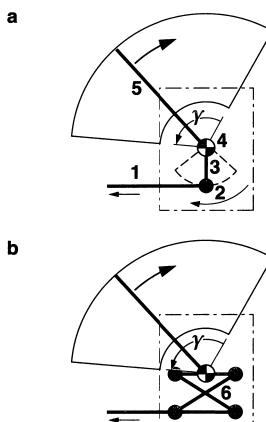
Wiper arms

The wiper arm is the connecting link between the wiper linkage and the wiper blade. It is screwed to the tapered wiper bearing shaft by its mounting end, which is usually made of diecast aluminum or steel sheet. The other end is generally a steel strip and holds the wiper blade (Figure 6). A distinction is made here between hook-type, side and front fastenings (Figure 7).

The wiper bearing is screwed to the body. The position of the wiper arm in relation to the windshield is established by way of the position of the wiper bearing relative to the windshield. The field of vision is established in combination with the wiper-blade length and the wiping angle.

Figure 5: Wiper-system linkage mechanism

- a) Conventional linkage,
 - b) Crosslay linkage.
 - 1 Articulated rod,
 - 2 Oscillating-crank link as ball pivot,
 - 3 Oscillating crank,
 - 4 Wiper bearing,
 - 5 Wiper arm,
 - 6 Crosslay.
 - γ Wiping angle.
- The arrows indicate the direction of motion.



In addition to the standard wiper-arm design, there are a number of design variations. These are also special designs of wiper arms, which perform, for example, the following additional functions:

Wiper arm with quadruple-link reciprocating action

The wiper arm with quadruple-link reciprocating action specifically shifts the wipe

pattern on the windshield, usually on the passenger side. This reduces the size of the unwiped area in the top corner of the windshield.

Wiper arm with wiper-blade control

The wiper arm with wiper-blade control provides an additional rotation of the wiper blade relative to the wiper arm, in order, for example, to wipe parallel to the A-pillar on wiper systems with only one wiper blade (Figure 1c).

Parallelogram wiper arm

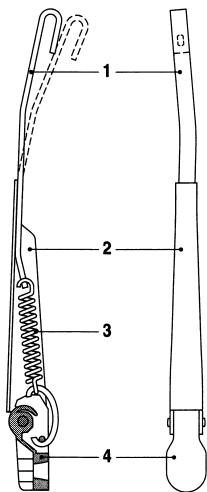
The parallelogram wiper arm is a special type with wiper-blade control. It holds the wiper blade in a fixed position over the entire wipe movement (e.g. in the vertical position on local-transport buses).

Wiper-blade positioning

A second optimization step involves the work position of the wiper-element lip relative to the surface of the windshield or rear window (see Wiper-blade element). The position of the wiper blades is defined by the angular position of the wiper bearings to the windshield and by applying additional torsion to the wiper arms. The aim is to incline the wiper arms laterally at their reversal points towards the wipe-angle bisector. This helps the wiper-blade elements to swivel into their new working position. This, in turn, reduces wiper-blade wear and the reversing noise.

Figure 6: Wiper arm (side and top views)

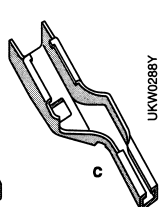
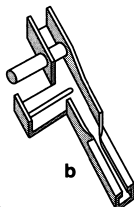
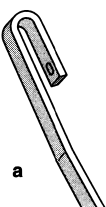
- 1 Steel strip with hook-type fastening,
- 2 Joint section,
- 3 Tension spring,
- 4 Attachment part with taper for mounting on wiper bearing shaft.



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Figure 7: Attachment of wiper blade to wiper arm

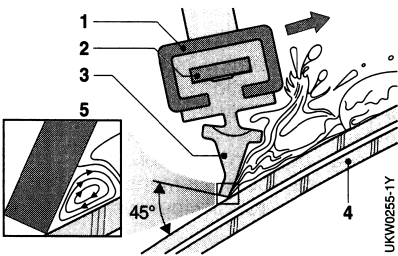
- a) Hook-type fastening,
- b) Side fastening,
- c) Front fastening.



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Figure 8: Wiper-blade element in working position

- 1 Claw,
- 2 Spring strip,
- 3 Wiper-element lip,
- 4 Windshield,
- 5 Double microedge.



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Wiper blades*Wiper-blade element*

The most important component of the wiper system is the rubber wiper-blade element. It is retained by the claws of the bracket system (Figure 9) or supported by spring strips. The edge of the wiper-element lip touches the windshield over a width of 0.01 to 0.015 mm. When moving across the windshield, the wiper-blade element must overcome coefficients of dry friction of 0.8 to 2.5 (depending on air humidity), and coefficients of wet friction of 0.6 to 0.1 (depending on frictional velocity). The correct combination of wiper-element profile and rubber properties must be chosen so that the wiper lip can cover the complete wipe pattern on the windshield surface at an angle of approx. 45° (Figure 8).

The twin wiper, with a two-component wiper-blade element made of synthetic

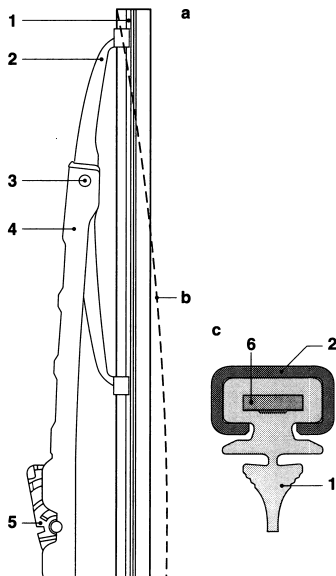
rubber, consists of a specially hardened, abrasion-resistant wiper-element lip which merges into an extra-soft spine. The soft spine ensures that the wiper element has optimum reversing characteristics and wipes smoothly.

Conventional wiper blade

The wiper blade (Figure 9) holds the wiper-blade element and directs its movement across the windshield. Wiper blades are used in lengths of 260 to 1,000 mm. Their dimensions for mounting (e.g. hook-type or snap-in fastening) are standardized. Low-wear operation is achieved by compensating the play in their mountings and linkages. The tops of the center brackets are perforated to prevent blade liftoff at high speeds. In special cases, wind deflectors are integrated into the wiper arms or blades to press the blades against the windshield.

Figure 9: Conventional wiper blade

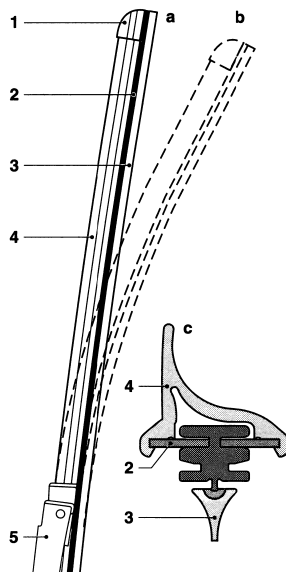
- a) Wiper blade under load, b) At zero load, c) Cross-section.
1 Wiper-blade element, 2 Claw bracket,
3 Joint, 4 Center bracket, 5 Adapter,
6 Spring strip.



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Figure 10: Flat wiper blade (aero-wiper blade)

- a) Wiper blade under load, b) At zero load, c) Cross-section.
1 End clip, 2 Spring strip,
3 Wiper-blade element,
4 Spoiler, 5 Wiper arm.



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Flat wiper blade (aero-wiper blade)

The flat wiper blade (aero-wiper blade) is the present trend in wiper-blade design (Figure 10). The contact pressure over the wiper-blade element is no longer distributed by the claws of the wiper bracket, but by two preshaped spring strips (sprung rails) specially adapted to the shape of the windshield. They press the wiper-element lip more evenly against the windshield. This reduces wear on the wiper-element lip and increases wipe quality. In addition, the elimination of the bracket system means that there is no linkage wear, the overall height of the wiper system is substantially reduced, the weight is lower, and the wiper is quieter (there is also less wind noise).

The top edge of the wiper blade is in the shape of a spoiler (air deflector) and allows the blade to be used even at very high speeds without any modification. The flexible material of the spoiler is also much less likely to cause injury to pedestrians in the event of an accident (pedestrian protection).

An adapted, simplified connection to the wiper arm ensures a reliable attachment of the wiper blade when the wipers are in operation, and also allow easy replacement when required.

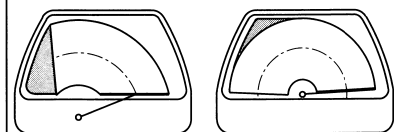
Rain sensor

The rain sensor (see chapter entitled "Sensors") detects how heavy the rain is, and converts this data into an appropriate signal that is sent to the wiper motor. The motor is then automatically switched on and set to intermittent mode or speed one or two as required.

The full potential of the rain sensor is fully realized with an electronically controlled wiper system, whose wiping speed can be infinitely re-adjusted to the rain intensity.

Figure 11: Rear-window wipe patterns

The gray areas represent impaired visibility when vehicles are overtaking.



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Rear-window wiper systems**Function and requirements**

Rear-window wiper systems are used when the rear window, on account of its sloping angle or the body shape, is prone to getting very dirty and rear visibility is compromised. The principle of rear-window cleaning is generally the same as that for windshield cleaning systems.

A rear-window cleaning system is subject to much lower demands than that of a windshield cleaning system. A rear-window cleaning system is therefore frequently operated in intermittent mode and its field of vision is not subject to any statutory requirements. The wiping angle typically ranges between 60° and 180° (Figure 11).

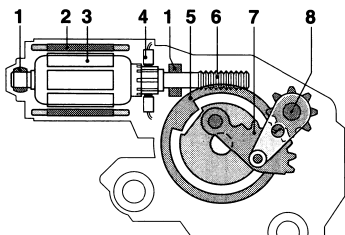
Rear drives

Rear drives are essentially the same as front drives. A rear-window wiper system is usually driven by an electric motor with integrated oscillating mechanism, which assumes the function of the linkage in windshield cleaning systems and performs the oscillating movement of the output shaft (Figure 12). The wiper arm is directly attached to the output shaft of the wiper drive.

To increase the wiping angle while at the same time taking up minimal space, solutions have been implemented whereby the classic four-bar mechanism increases the oscillation angle to a value up to 180° by means of an additional rotational relative movement.

Figure 12: Rear drive with oscillating mechanism

- 1 Bearing, 2 Magnet, 3 Armature,
- 4 Commutator with carbon brushes,
- 5 Helical gear, 6 Worm,
- 7 Oscillating mechanism, 8 Output shaft.



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Windshield and rear-window washer systems

Body

To ensure good visibility in the wipe pattern, it is imperative that the wiper system is backed by a washer system. Electrically driven centrifugal pumps are used to direct water mixed with detergent additive from the washer-fluid reservoir through 2–4 nozzles in a pointed spray or through sprinkler nozzles as water mist onto the windshield (Figure 13). The capacity of the washer-fluid reservoir is usually 1.5 to 2 l. If the same washer-fluid reservoir is also used to supply the headlamp washers, a capacity of up to 7 l may be required. A separate reservoir may be provided for the rear-window cleaning system.

Electronic control

The washer system is often linked to the cleaning system by means of an electronic control system so that water is sprayed

onto the rear window or windshield for as long as a pushbutton remains pressed. The wiper system then continues to operate for several additional cycles after the pushbutton is released.

Headlamp washer systems

Pure washer systems have become established for cleaning headlamps. The advantages of the headlamp washer system over the previously used wipe/wash systems lie in its simple design and in the fact that it is easier to adapt to the vehicle styling concept.

A headlamp washer system is prescribed by law in Germany for xenon headlamps to prevent oncoming traffic from being dazzled by light diffusion.

Body

High-pressure washer systems (Figure 13) consist of the washer-fluid reservoir (the washer fluid required is taken from the reservoir for the windshield washer system), the pump, tubes with a non-return valve, and the nozzle holders (horn) with one or more nozzles. As well as fixed nozzle holders on the bumpers, there are also telescopically extending nozzle holders. The telescope improves the cleaning effect because it can assume an optimum spray position. In addition, when not in use, the nozzle holder can be concealed, e.g. inside the bumper.

Cleaning effect

The cleaning effect is chiefly determined by the cleaning pulse of the water droplets. The decisive factors here are the distance between the nozzles and the lens, the size, the contact angle, the contact velocity of the washer-fluid droplets, and the amount of washer fluid. It is important that the nozzles be positioned correctly so that water jets properly cover the headlamps at all driving speeds.

References

- [1] Council Directive 78/318/EEC: Wiper and washer systems of motor vehicles.
- [2] FMVSS Part 571, Standard No. 104: Windshield Wiping and Washing Systems.
- [3] SAE J903: Passenger Car Windshield Wiper Systems.

